JUNE 1961

experimental mechanics

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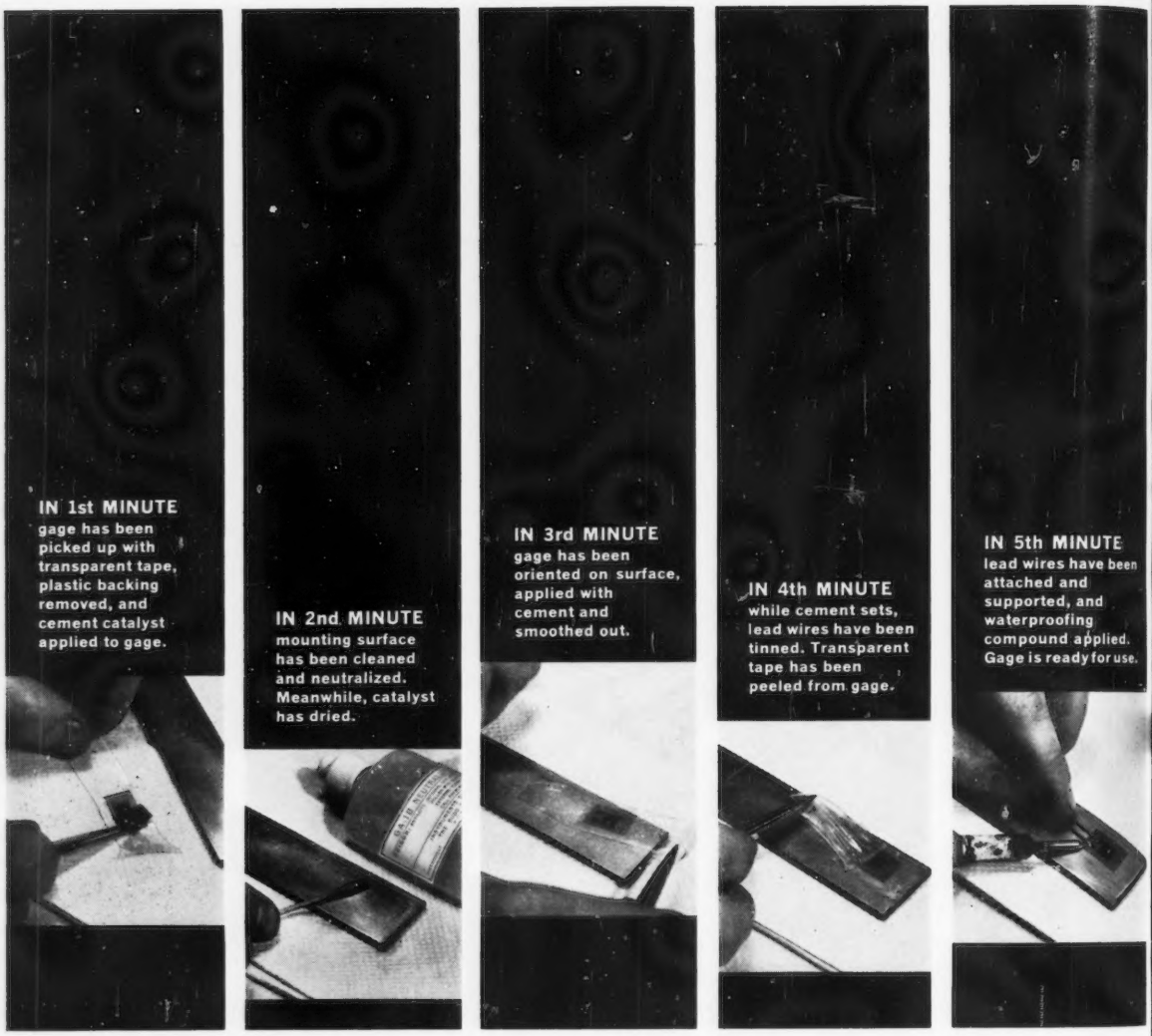
SESA JOURNAL OF THE SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS


IN 1st MINUTE
gage has been picked up with transparent tape, plastic backing removed, and cement catalyst applied to gage.
IN 2nd MINUTE
mounting surface has been cleaned and neutralized. Meanwhile, catalyst has dried.
IN 3rd MINUTE
gage has been oriented on surface, applied with cement and smoothed out.
IN 4th MINUTE
while cement sets, lead wires have been tinned. Transparent tape has been peeled from gage.
IN 5th MINUTE
lead wires have been attached and supported, and waterproofing compound applied. Gage is ready for use.

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SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS
21 Bridge Square, Westport, Connecticut

SESA PUBLICATIONS

PROCEEDINGS OF THE SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS

The PROCEEDINGS are a series of bound volumes comprised of the technical papers presented at National Meetings. These contain the results and descriptions of analytical and experimental investigations, as well as techniques for the measurement of mechanical strain. Published twice annually, they include up-to-date information on various phases of instrumentation. Currently available:

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CUMULATIVE INDEX—PROCEEDINGS

Thirty-four books of the PROCEEDINGS have been published to date. This Index has been prepared to serve as an easy reference to any one of the 546 papers which have been published in the first thirty books. It consists of three parts: (1) chronological index, (2) author index, (3) topical index. It represents a "master reference file" to all PROCEEDINGS published, from Vol. I, No. 1 through Vol. XV, No. 2.

HANDBOOK OF EXPERIMENTAL STRESS ANALYSIS

This book opens with a thorough presentation of mechanical properties. It presents practical, usable information on: mechanical, electrical and optical gages, brittle coatings, photoelasticity, X-rays, service fractures, analogies, vibration measurements, theory of elasticity and precision of measurements. Stiff cloth binding—contains 1060 pages of text.

MANUAL ON EXPERIMENTAL STRESS ANALYSIS TECHNIQUES

For those who are new to the field of experimental stress analysis, the MANUAL introduces the fundamentals of experimental stress analysis in terms easily understood by the novice. It contains detailed instructions and, with the aid of illustrations and charts, elaborates on the application of three particular methods for determining stress: (1) SR-4 bonded wire strain gages, (2) stresscoat (brittle lacquer) and (3) photoelasticity.

STRAIN GAGE TECHNIQUES

Twenty-three chapters of lectures and laboratory experiments of special summer programs presented at leading universities, covering such topics as: fundamental concepts, selection of gages, basic considerations in strain-gage instrumentation, electrical and mechanical aspects of the gage system, the potentiometric circuit, the wheatstone bridge, amplitude modulation, fundamentals of measuring systems, some commercial systems and rosette analysis. Approximately 680 pages.

CATALOG OF EQUIPMENT USED IN EXPERIMENTAL STRESS ANALYSIS

The Catalog contains references to manufacturers of equipment, hardware and instrumentation used in the measurement of strain.

EXPERIMENTAL MECHANICS

Official monthly journal of the SESA. Includes technical papers and news items of interest to SESA members and to industry.

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Experimentally Speaking

KNOWING IS NOT ENOUGH

There is a big difference between knowledge and understanding. A brain crammed with knowledge is of relatively small value if the bits of knowledge cannot be called forth and properly transmitted to others. Engineers, particularly young ones, sometimes become frustrated because good ideas are not picked up by their supervisors and put to work. More than likely, failure to pick up the idea results from distortion or lack of clarity in transmission from the one to the other. There are at least two reasons for situations of this kind.

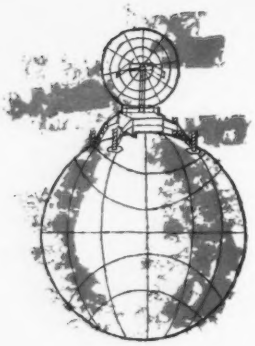
Mankind has more ideas than words to express them so that some words are associated with several ideas. In fact, my dictionary gives ten uses for "word." On the other hand, there are many groups of words which have almost the same meaning; for example, one might use any of the following verbs to indicate that some new thing or idea has been brought to light: reveal, show, manifest, exhibit, display, disclose, evidence, prove and divulge. The manner in which a new thing is developed can be intimated by the choice of the verb. It is thus apparent that the choice of the proper word is very necessary for the clear and facile transmission of a thought from one mind to another. A poor selection of words can be covered up by the use of additional words, but these additional words extend the time required for the transmission of the idea and they require further effort on the part of the receiver to grasp the idea. In addition to having knowledge of a subject, the teller must have such an understanding that the correct words can be chosen to create in the minds of others the image of the idea in his mind.

Many speakers and writers are careless or ignorant of the rules of good grammar. No serious reader should be required to expend effort to ferret out even a good idea from a mass of jumbled, mutilated sentences. Correct punctuation will often clarify the meaning of a word, and indicate whether it is used as an adjective or as a noun.

Effective speaking and writing are not accidents; they result from careful attention to details. Pleasant inflection of the voice can be used to emphasize key points and will help in keeping a listener's interest awake. Clear sketches and pictures may save many words.

Knowing is not enough; to be successful, one must make it easy for others to know, too.

Marshall Holt
PAST PRESIDENT, SESA



tracking station

the industry in focus

Nuclear-powered Rocket Propulsion

A more efficient system for nuclear-powered space-vehicle propulsion involving super-heated hydrogen as a propellant has been proposed by two aeronautical engineers at Cornell University as an alternative to the chemical units now in use. The hydrogen and helium in their nuclear power plant scheme provides higher specific impulse than before, according to Edwin L. Resler, Jr. and Nicholas Rott.

Chemical burning rockets such as those being used today are incapable of achieving high speeds with which to conquer light-years of space. By today's standards a specific impulse of 300 sec indicates an extremely good fuel. The new system is said to provide 750 sec specific impulse—even after making allowances for probable power losses.

World's Tallest Meteorological Structure

A cloud-scraping television tower near Dallas, Tex., is now doing double duty as the world's tallest meteorological research structure, the USAF Air Research & Development Command announced recently. Rising 1428 ft above the Texas plains, the structure contains instruments to automatically measure and record winds and temperatures.

The tower is jointly owned by stations WFAA-TV and KRLD-TV. Part of the cost of equipping the tower was contributed by the Army Signal Corps. Principal object of this project is to investigate a weather phenomenon known as the "low-level jet," a sharp maximum of wind speed at an average height of 1000 and 1500 ft above the earth's surface. At the level of the maximum, the wind speed often reaches 50 or 60 mph, accompanied by much lower values both below and above this level.

The jet develops after sunset, gradually increases in intensity until a few hours after midnight, and then shows a gradual decay until shortly after sunrise.

Oxygen Retrieved from Astronaut's Breath

A major step in developing a system to convert an astronaut's breath into breathable oxygen has been reported by Battelle Institute.

A complete oxygen-recovery system consists of three parts: a device to recover exhaled carbon dioxide from the cabin of a space vehicle; a device to convert the carbon dioxide into carbon and water; a device to electrolyze the water into oxygen and hydrogen.

Research sponsored by the Air Research and Development Command, has resulted in a working prototype of the device which converts carbon dioxide into carbon and water. It is designed to operate on space voyages lasting as long as three years.

Freshman Engineering Enrollments Drop

Engineering freshmen enrollments have dropped for the third consecutive year, notwithstanding preliminary indications of a possible upturn. Registrations for the 1960–61 academic year were only about 100 less than the previous year's 67,700 according to an Office of Education survey, but the significance lies in the 12.4% rise in total college freshmen enrollments during the same interval. The proportion of engineering freshmen to total freshmen enrollment has declined steadily from 10.8% in 1957 to 7.3% in 1960.

The 234,200 total engineering undergraduates are 3.7% lower than for the fall of 1959, but graduate enrollments continue to rise: masters 5.1% (31,200) and doctors 14.3% (6,400).

Fairchild Reports 1960 Losses

Fairchild Engine and Airplane Corp. announced a 1960 net loss of $5,994,650 on sales and other income of $85,774,153. Earnings in 1959 were $1,515,261 on sales of $114,068,426. Backlog at the close of 1960 was $43 million as against $76 million at the close of 1959. Loss per share in 1960 amounted to $1.97 as against an earning per share of 50 cents in 1959.

Virtually all of the 1960 loss resulted from a downward revaluation of F-27 inventories and a recording in 1960 of anticipated losses on uncompleted government subcontracts.

Fairchild Engine and Airplane Corp. is a corporation member of SESA.

FUTURE MEETINGS

SESA

NOVEMBER 1–3. First International Congress on Experimental Mechanics. Hotel New Yorker, New York, N. Y.

1962 Spring Meeting, Dallas, Texas

• Related Conference:

OCTOBER 29–31. International Symposium on Photoelasticity. Illinois Institute of Technology, Chicago, Ill.

ASME

JUNE 14–16. Conference of American Society of Mechanical Engineers. Applied Mechanics Conference. Illinois Institute of Technology, Chicago, Ill.

ASTM

JUNE 25–30. Annual Meeting of American Society for Testing Materials. Chalfonte-Haddon Hall, Atlantic City, N. J.

NRL

OCTOBER 10–12. The 30th Symposium on Shock, Vibration and Associated Environments. Detroit, Mich. Classified. Address: Code 4021 U. S. Naval Research Laboratory, Washington, 25 , D. C.

OSR

JULY 31–AUGUST 4. Conference of Air Force Office of Scientific Research and Martin's Research Institute for Advanced Studies. Differential Equations in Nonlinear Mechanics. Air Force Academy, Colorado Springs, Colo.

SAE

JUNE 4–9. Summer Meeting of the Society of Automotive Engineers. Chase-Park Plaza Hotel, St. Louis, Mo.

personnel *in the news*

JASPER NAMED CHIEF SCIENTIST

Norman H. Jasper left the David Taylor Model Basin at the end of March after fifteen years of increasing technical responsibility to go to the U. S. Navy Mine Defense Laboratory as its chief scientist. Dr. Jasper came to the Taylor Model Basin in February 1946 as a physicist and general engineer in the Vibrations Division of the Structural Mechanics Laboratory. He became head of the Applied Dynamics Branch in 1954 and successively was head of the Vibration and Seaworthiness Branch and of the Ship Dynamics Division. His most recent assignment was coordinator for a specific noise reduction program for the Bureau of Ships, reporting directly to the commanding officer and director of the Taylor Model Basin.

Prior to coming to the Taylor Model Basin, Dr. Jasper spent five years as a naval architect in the Scientific Section of the Puget Sound Naval Shipyard.

In 1941 he received a B.M.E. degree from College of the City of New York. The University of Maryland granted him an M.S. degree in 1952 and in 1956 he obtained his Doctor of Engineering degree (Aeronautical Engineering) from the Catholic University of America.

In the field of structural seaworthiness he initiated and developed a statistical approach to the strength design of ships, and developed automatic statistical recorders which enabled the Taylor Model Basin and others to obtain a large collection of data on strains and motions in ships at sea and to correlate these with environmental conditions. He has studied and developed structural design methods for high-speed planing craft. He recently directed an analog-computer program which permits a study of the ship's dynamic response to specified sea conditions. His investigations in ship vibrations include a detailed study of ship propeller shaft failures which has resulted in modifications of shaft design practice.

Dr. Jasper has been a member for many years of the Society for Experimental Stress Analysis.

BARNABY BECOMES R & D MANAGER

Ralph S. Barnaby, Captain (USN Ret.), a staff member of The Franklin Institute, has been named manager of Project Development of The Franklin Institute Laboratories for Research and Development.

He will assist Francis L. Jackson, assistant director of the Laboratories, in planning projects for the 300 scientists and engineers in the fields of physics, chemistry, engineering psychology and operations analysis, and in developing production, power, transportation and satellite systems.

Capt. Barnaby, formerly principal engineer in the Operations Analysis Laboratory, has been an Institute research staff member since 1947. He came to the Institute after completing a 30-year career in Naval aviation, during which he distinguished himself as an aviator, engineer and administrator.

CENTER MADE GENERAL MANAGER

Maurice Center, until recently program director for Martin's Missile Master system, has been named general manager of Fairchild Astrionics Division at Wyandanch, Long Island. Mr. Center succeeds Grayson Merrill, general manager of the division since 1958, who recently resigned to accept another position in industry.

A 1947 graduate of Purdue University, with a degree in electrical engineering, Mr. Center also studied contract law at Johns Hopkins University.

Mr. Center joined The Martin Co. in 1951 as an electronic design engineer.

COOPER JOINS MICRO SYSTEMS

Theodore W. Cooper has joined the staff of Micro Systems, Inc., as manager of the organization's Operations Division. In this position, Mr. Cooper will be in charge of manufacturing the company's product lines, as well as such supporting functions as production control, quality control and product development.

Prior to joining Micro Systems, Mr. Cooper was manager of Equipment Engineering for the Semiconductor Division of Hughes Aircraft Co. He was previously head of Development Engineering at that company, and was responsible for product, process and equipment development for semiconductor diodes and transistors.

He received his B.S. and M.S. degrees in mechanical engineering from Michigan College of Mining and Technology.

Mr. Cooper holds many patents in the field of semiconductors, having contributed primarily to transistor product and process designs.

READ ELECTED PRESIDENT

Ralph C. Read has been elected president of Cenco Instruments Corp. it was announced by Alfred A. Strelsin, chairman and chief executive officer.

Mr. Read, an industrial engineer with a degree from the University of Michigan, joined Cenco in 1956 as manager of manufacturing and became executive vice president in 1959. As president he succeeds Mr. Strelsin who became chairman of the board.

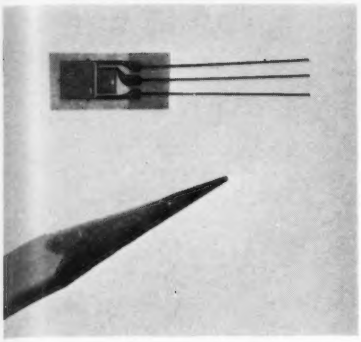
Mr. Read was also elected a director of the company to succeed William F. Crawford who resigned because of the pressure of his own business.

Cenco Instruments, has plants in Chicago, Ill., Berkeley, Calif., and in Europe.

EMPLOYMENT SERVICE BULLETIN

Services Available

A-1. ESA Engineering. BSME. One year graduate work including studies on surface photoelasticity. Age 29. Two years work with bondable resistance-type strain gages and related instrumentation for studying stress and vibration on turbine blades. Trouble shooting and operation of automatic digital printing instrumentation for multigage testing. One year work with strain gages and stress coat in aircraft testing. High-speed slip ring design. Test fixture design. Prefer West. Available end of semester, June 1.



STRAIN GAGE FOR STRAIN-STRESS EQUATION

A bonded resistance-foil strain gage—with a built-in computer that solves general strain-to-stress equations automatically—has been developed by Electronics & Instrumentation Division, Waltham, Mass., of Baldwin-Lima-Hamilton Corp.

It is reported that the new gage will simplify the task of obtaining stress readings in a wide range of testing and measuring applications by automatically eliminating the need for tedious, time-consuming calculations of stresses from the strain indications.

The new SR-4 stress-strain gage is said to provide electrical responses which are proportional to either stress or strain, at the discretion of the user, by using two independent axial strain-sensing elements oriented 90 deg apart.

One element measures the conventional strain. The other element acts as the automatic computer by rejecting the axial component of strain caused by stress in a transverse direction. The combined elements then respond only to that component of strain which is produced by stress in the axial direction.

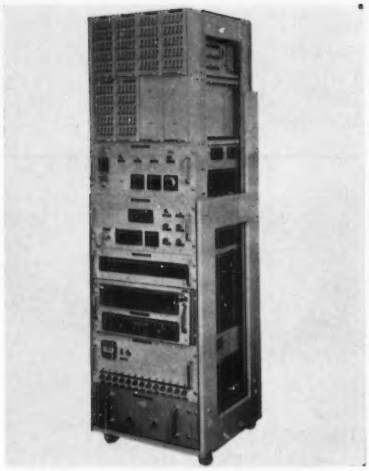
The two sensing elements have a common electrical connection to permit independent use for measuring conventional axial or transverse strains, or combined use to measure the stress along the principal gage axis.

The new gages are available with a phenolic base construction and constantan foil—with resistance ratios and temperature compensation for mild steels, stainless steel and aluminum. **51**

STRAIN-GAGE DATA LOGGER

Their Model 179 strain-gage logger, a 200-channel instrument contained in one standard 19-in. relay rack, has been announced by Gilmore Industries, Inc., 13015 Woodland Ave., Cleveland 20, Ohio.

The Model 179 is described as a medium speed, absolute millivolt instrument, which scans 200 channels of strain data at a rate of 5 channels per second. In addition to printing these data out on paper tape, the instrument also is said to have a high-speed tape-punch system which provides a coded punched tape record of all data. This information can be fed to a tape reader and/or computer for further analyses and plotting of stress-strain graphs. Either of two



full-scale strain ranges, zero to 5000, or zero to 10,000 microin./in., can be selected by a convenient front panel switch. The unit also has an alarm system, which will indicate when any given input exceeds a preset strain value for any channel.

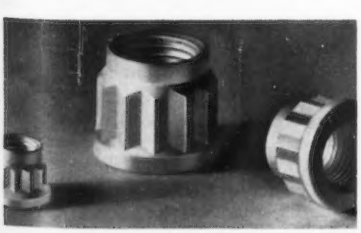
With the exception of the printer and tape punch, the entire instrument is contained, in one, shock-mounted, 6-ft by 19-in. relay rack. **53**

new products

For more details regarding these products, circle item number on Reader Information Card

HIGH-TEMPERATURE LOCKNUT

An elevated-temperature locknut made of Waspalloy, a nickel-base alloy,



is intended for use with high performance bolting up to 1400° F.

Designated FN 1418, the one-piece self-locking nut, developed by Standard Pressed Steel Co., Jenkintown, Pa., has properties which reportedly make it possible to utilize the full potential of high-strength 1200 and 1400° F bolts.

Principal applications of the new locknut are on jet and rocket engines, missiles, gas turbines and other high-temperature power generation equipment. **52**

SCOPE-MOBILE CART

The Tektronix Type 200-Series Scope-Mobile carts are said to provide a convenient mobile support for their oscilloscopes or auxiliary equipment according to Tektronix, Inc., P. O. Box 500, Beaverton, Ore. They feature an easily adjustable tray which can be tilt-locked in any one of nine positions. They also feature an optional plug-in carrier which can hold additional plug-ins.

The Type 201—with tray width of $10^1/_2$ in.—accommodates medium-size oscilloscopes. Type 202—with tray width of 14 in.—accommodates large-size oscilloscopes. Dimensions for both Scope-Mobile carts are approximately 35 x $17^1/_2$ x 27 in. deep. Weight is 35 lb. **54**

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TORQUE-WRENCH TESTER

Bytrex Corp., 50 Hunt St., Newton 58, Mass., announces a new torque-wrench tester for power-operated torque-controlled tools which is said to be also very well suited to calibrating hand torque wrenches. The manufacturer guarantees accuracy compatible with the best wrenches and states that the unit conforms to military specifications. The tester covers the range of 0–1200 in.-lb and is suitable for use with wrench spindle speeds up to 500 rpm. A set of adapters for various drives is housed inside the unit as well as a calibration arm with weights to provide 5 point calibration on all ranges. **55**

FILM ADHESIVE

A structural thermosetting film adhesive with unique curing properties for metal-to-metal bonding and metal honeycomb sandwich construction; contact bond adhesives for industrial applications; and a variety of adhesives for bonding copper foil in printed circuit production are announced by Minnesota Mining and Manufacturing Co., 900 Bush Ave., St. Paul 6, Minn.

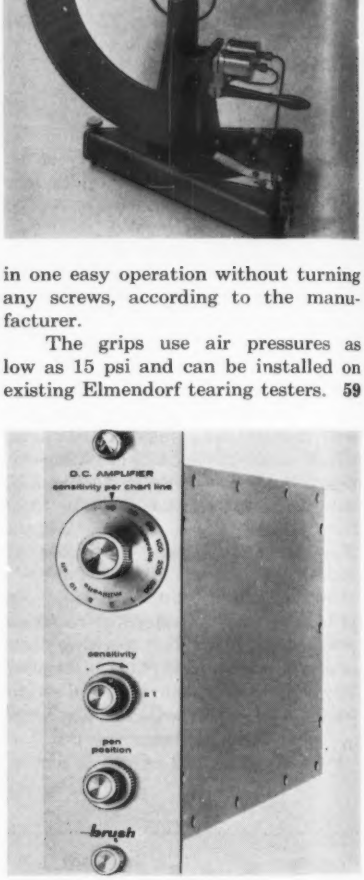
According to the manufacturer, the new 3M film adhesive, designated as AF-110, can be used for low-pressure bonding, requiring only enough pressure during the curing operation to provide contact between the parts being bonded and to keep them properly aligned. **56**

AIR GRIPS FOR TEARING TESTER

Thwing-Albert Instrument Co., Penn St. and Pulaski Ave., Philadelphia 44, Pa., announces the development of air-operated grips for the Elmendorf tearing tester. The grips reportedly enable an operator to test twice as many samples in less time with greatly reduced operator fatigue. Samples can be inserted and removed

in one easy operation without turning any screws, according to the manufacturer.

The grips use air pressures as low as 15 psi and can be installed on existing Elmendorf tearing testers. **59**



TRANSISTORIZED PREAMPLIFIER

A compact, transistorized plug-in preamplifier designed to measure signals as low as 10 microvolts is available from Brush Instruments, division of Clevite Corp., 37th and Perkins, Cleveland 14, Ohio. Designated RD 4215 20 Very High Gain Preamplifier, the unit is one of the company's new 4200 Series of transistorized preamplifiers which adapt Brush or other medium-gain recording systems to a wider range of recording applications.

According to the manufacturer, the chopper-stabilized preamplifier has a sensitivity of 10 microvolts per chart line in direct-writing recorder applications. The 10,000 ohm input impedance of the unit is floating and guarded. Also available is a high-gain amplifier, RD 4215 00, which has a 1 megohm input and a sensitivity of 100 microvolts per chart line in direct-writing recorder applications. **60**

STRUCTURES ENGINEERS

Inquire about new opportunities for highly qualified engineers in the fields of

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Atlanta 8, Georgia

LOCKHEED/GEORGIA

All qualified applicants will receive consideration for employment without regard to race, creed, color or national origin.

PRESSURE TRANSDUCER

A versatile general-purpose zero-length/dimensionless unbonded strain gage pressure transducer, the P707TC, has been introduced by Statham Instruments, Inc., 12401 W. Olympic Blvd., Los Angeles 64, Calif. Available for measurement of absolute and gage pressures from 0 to 5 psi through 0–5000 psi, the P707TC has been designed to withstand the stringent environmental conditions encountered in high-performance aircraft, space probes and ballistic missiles, according to the manufacturer.

Other features of this pressure transducer are said to be its small size, light weight, low sensitivity to acceleration, and excellent thermal performance without sacrifice in frequency response or in nonlinearity and hysteresis characteristics.

Excitation of the P707TC is 5 v (for gage model) and 7 v (for absolute model), dc or ac (rms) through carrier frequencies. Output of the gage model is 25 mv, and output for the absolute model is 35 mv, full-scale open circuit. Ambient temperature limits are $-100°$ F to $+275°$ F. The combined error due to nonlinearity and hysteresis is less than $\pm 0.75\%$ of full-scale output, it is stated. **61**

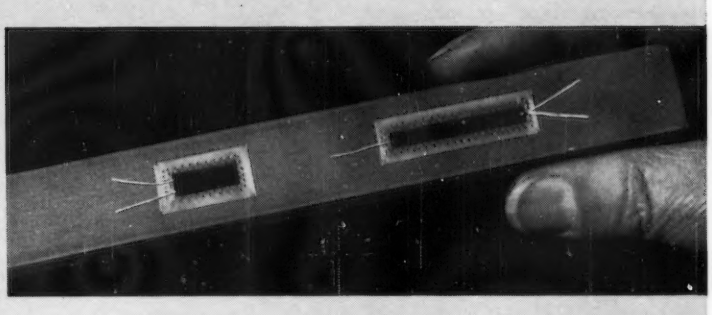
AUTOMATIC CARD READER

A card reader, designed for use with card-programmed industrial control systems, is available from Datex Corp., 1307 S. Myrtle Ave., Monrovia, Calif.

Called the CR-201, it is said to transport IBM cards from a card hopper past two read stations and into a card stacker at a speed of 30 cards per minute. The cards are scanned row by row and contact closure outputs are provided corresponding to the perforations in all 80 columns of both cards. Rows may be selected and reselected in any order as required by control routines. Scanning is independent of card feed and, if required, cards may be scanned many times before a new card is advanced.

The manufacturer also states that the card reader may be operated either from an integral control panel or by a remote automatic control system. Three modes of operation are available—continuous scanning, automatic stop at any required row and continuous readout and individual selection of any required row. **62**

BLH ADVANCES THE STRAIN GAGE ART WITH NEW WELDABLE, PRE-STABILIZED GAGES THAT INSTALL IN MINUTES



For the first time, strain gages that are both easily weldable and electrically stable from cryogenic through 750°F temperatures.

New BLH Weldable Strain Gages incorporate special SR-4® foil gages bonded to stainless steel shim stock .005 in. thick which can be matched to the coefficient of the test material. This new development enables gages to be closer to the test surface, improves conformity with the surface, and eliminates local buckling.

No curing needed after installation, because the gages are fully stabilized by heat treating. They offer high linearity of strain response in either tension or compression and operate in a strain range of ±0.5%.

Two types available—a phenolic-bonded Constantan foil for temperatures to 500°F, and a ceramic-bonded Nichrome V foil for temperatures to 750°F and higher. Temperature compensation is available to ±1 ppm/°F for units utilizing Constantan foil.

Wide range of applications is seen for testing nuclear reactor power loops, liquefied gas pressure vessels and systems, rocket and missile components and engines, and wind tunnel and shock chamber testing of aircraft parts.

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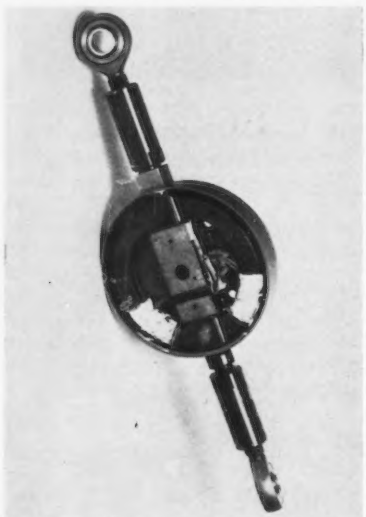
MICRODOT ANNOUNCES NEW FORCE MEASURING TRANSDUCERS FOR LOADS TO 500,000 LBS.

Microdot's ring configuration load cells measure both tension and compression loads over a full scale range of 100 to 500,000 lbs. Their sensing function is performed by Microdot's resistance wire-type high temperature weldable strain gages in a bridge configuration. An added link in the center of the ring allows considerable overload without destroying calibration.



Particular design attention has been given to temperature stability allowing for cases where there is extreme temperature gradient across the transducer. The weldable strain gages are compensated to have a low temperature sensitivity when welded to the base metal of the ring. Moreover, these gages contain no organic materials and have been successfully employed in radiation environments. Bridge impedance of the transducer is 120 ohm. Recommended input voltage is 10 v., with a maximum of 15 v. allowable. Output voltage is 25 mv. full scale for 10 v. input. Linearity and hysteresis are both within 0.2%. Total error from all causes is less than ±1% over the temperature range of −300° to +750°F. Deflection is approximately 0.010″ at full rated load. The no-load output of ±1% of full load output can be balanced out with usual balance circuits. The units can be provided sealed and are inherently rugged.

MICRODOT INC.

220 Pasadena Avenue
South Pasadena, California

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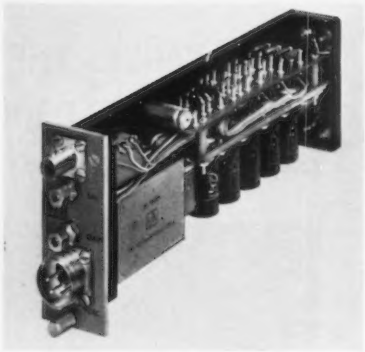


INSTRUMENTATION SYSTEM

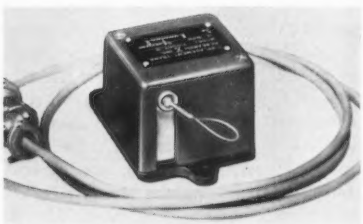
Model 96 RC is a miniaturized instrumentation system offered by Video Instruments Co., Inc., 3002 Pennsylvania Ave., Santa Monica, Calif., who describe it as follows: Slightly larger than a pack of king size cigarettes, the 96 RC contains a signal amplifier, transducer power supply, bridge balance circuits and calibration circuits. The calibration circuits can be programmed from a remote source. Four reference points are provided, and the polarity of the calibration can be reversed to simulate the compression or tension of a strain gage. The system can be used with any bridge type transducer having one, two or four active arms. Transducer signals, typically in the millivolt region, are amplified to a maximum output of ±5 v dc. The signal level is suitable for telemetering applications; due to the relatively low output impedance (350 ohms) the 96 RC is also capable of driving high sensitivity galvanometers directly. Ruggedized construction and the choice of aircraft type relays permits use of the 96 RC for aircraft and mobile applications. **63**

HIGH-TEMPERATURE LEAD WIRE

Strain Gage Accessories Co., P.O. Box 281, Northridge, Calif., announces a special strain gage lead wire which can be exposed to temperatures as high as 2000° F. The new concept is in the insulation, which is double-wrapped over the lead wire and bound with a stainless-steel braid which also acts as a shield. This insulation is available applied over No. 28 Oxalloy or over thermocouple wires. The insulation remains intact and completely flexible even after exposure to temperatures of 1500° F for as long as 72 hr, it is reported. The resistance between the lead wire and the shield is in the hundreds of megohms after the same exposure. **64**

SYNCHRO AND RESOLVER RESISTANCE BRIDGES

Kearfott Division, General Precision, Inc., Little Falls, N. J., has added high accuracy synchro and resolver resistance bridges to its line of standard test equipment for both military and industrial applications. These bridges are described as consisting mainly of a balanced, noninductive resistance network—a delta arrangement for the synchro bridge and a four-sided arrangement for the resolver bridge. Tapped arms of both bridges provide voltage ratios calculated in terms of an errorless component. Finger-tip control is said to assure maximum ease of operation, and each bridge can be indexed to any multiple of 5 deg throughout a range of 0 to 360 deg. Designed for modular application, these units are reportedly compact and rugged, and qualify according to MIL and SAE specifications. **65**



DISPLACEMENT TRANSDUCER

Measuring the initial separation rate of missile or rocket parts in flight is the function of a special displacement transducer developed by Research, Inc., 2809 Wayzata Blvd., Minneapolis, Minn.

Described as basically similar to devices offered by the company for measuring specimen deflections in aircraft structural loading tests, it contains a precision multiturn potentiometer linked to a grooved reel on which a length of flexible cable is wound. As the missile part to which the cable end is attached separates from the part on which the reel assembly is mounted, the reel rotates and turns the potentiometer. A constant restraining torque is applied to the reel by a "Neg'ator" spring so that the cable is constantly loaded with 1 lb tension as it is withdrawn.

The unit has a separable cable with a ball at its end which is held by a notch in the reel groove. When the cable is fully extended (12 in.) it falls completely free of the reel, releasing all restraint. **66**

LOW-NOISE BATTERY SUBSTITUTE

The A4S series of Isoplys (isolated power supplies) are offered as low-noise battery substitutes by Elcor, Inc., Falls Church, Va. The high degree of isolation achieved with these supplies is claimed to make them well suited for use in bridge circuits, bootstrapped circuits, floating bias or reference circuits and direct-coupled d-c amplifier circuits where previously it was thought that only batteries would have the necessary isolation qualities. The noise introduced by these units into the circuits to which they are connected is said to be less than 10 microvolts peak-to-peak per kilohm impedance to ground.

These units are typified by low distributed capacitance (20pf) between secondary circuit (d-c output) and ground (consisting of core, case and primary shield). **67**

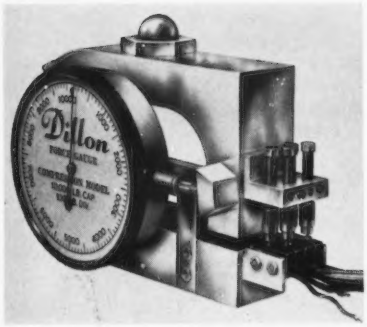
RF SWITCHING TECHNIQUE

The development of a ferrite RF switch intended for 225 to 400 mc application in rapid high-power antenna switching, lobe-shifting and time-division multiplexing is announced by Cook Electric Co., Chicago, Ill., and Adams-Russel Co., Inc., Cambridge, Mass.

The device makes use of a temperature-stabilized, low-loss ferrite ring whose electrical properties can be changed almost instantly by the sudden applications of a magnetic bias field. A set of such rings, appropriately arranged in a short coaxial line, forms a highly effective RF switch, entirely nonmechanical in operation and hence of quasi-infinite switching life. **68**

MICROSWITCH FORCE GAGE

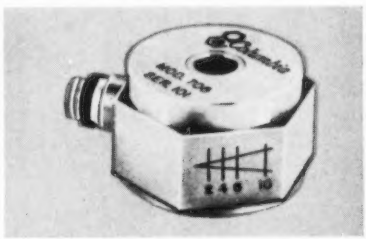
Presetting of control switches to trigger motors, bells, lights, etc., at desired load points, without the use of specialized calibrating equipment, is



possible with a new instrument offered by W. C. Dillon & Co., Van Nuys, Calif. It is available with up to 4 microswitches. Direct, visual indication is also provided by the force gage dial.

Typical applications according to the manufacturer are found on automated production lines where this instrument is set for permanent operation. Also, in R & D laboratories where there is a need to check operational characteristics of equipment when varying mechanical forces are applied. Changing loads will not only control job functions but may also be visually read.

Dillon microswitch force gages are available with 1, 2, 3 or 4 microswitches in compression, tension or push-pull models. Twelve capacities from 0 to 10 lb through 0–50,000 lb make up the complete line. **69**



TRUE-COMPRESSION ACCELEROMETER

The Model 706 accelerometer, is introduced as a true compression-sensing device for measuring shock and vibration in missile and airborne vehicles and in standard laboratory acceleration-measuring systems, by Columbia Research Laboratories, Mac Dade Blvd. & Bullens Lane, Woodlyn, Pa.

The manufacturer states that Model 706 is a general purpose unit featuring simple bolt-down mounting through a center clearance hole. This makes possible rotation of the output connector to any convenient direction in the plane perpendicular to its sensitive axis. Electrical isolation of the accelerometer case from ground is accomplished by use of insulating washers supplied with each unit. **70**

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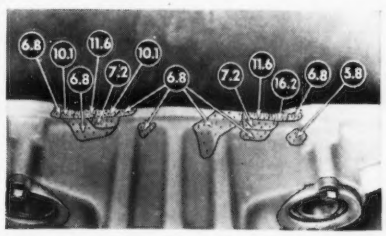

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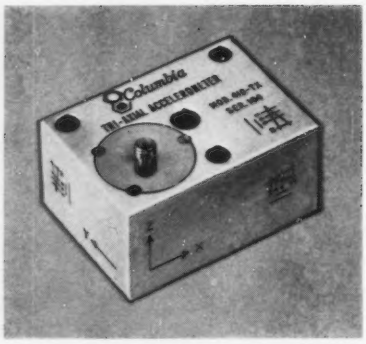


TRIAXIAL ACCELEROMETER

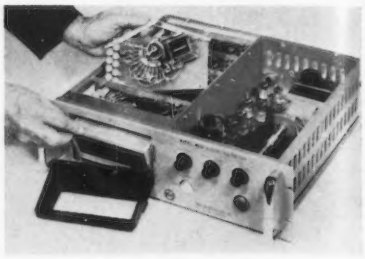
A triaxial accelerometer, Model 410-TX, a true compression-sensing device with a high sensitivity of 75 mv/g along its three mutually perpendicular planes, has been developed by Columbia Research Laboratories, MacDade Blvd. and Bullens Lane, Woodlyn, Pa. Applications for the Model 410-TX include low-level g studies of large turbines, ground and submerged vehicles, and other heavy equipment where high signal-to-noise ratio and low frequency response are important.

According to the manufacturer, acceleration range of the Model 410-TX is from 0.01 g to 1000 g with a maximum cross sensitivity of 5%. Frequency response is flat within ±5% from 0.2 cps to 3 kc, and resonant frequency is 20 kc. Response to acoustic noise is less than 0.1 g at 155 db. Open circuit sensitivity is 90 mv/g, amplitude linearity is ±1% and nominal capacity is 450 $\mu\mu$f. Temperature range is from −65 to +300° F; high temperature units go up to +500° F. **71**

SIZE 5 MOTOR GENERATOR

Helipot Division of Beckman Instruments, Inc., 2500 Harbor Blvd., Fullerton, Calif., announces initial production of a Size 5 servomotor-generator. Designed for applications where minimum size and weight are required, this precision control component weighs 1.1 oz, measures only 1.565 in. in length.

Motor and generator are contained in a single stainless-steel housing, and rotate on a common shaft. Excitation voltage for both motor and generator is 26 v, 400 cycles. Motor specs show torque at stall of 0.1 oz in., acceleration at stall, 37,000 rad/sec², while no-load speed developed is 10,000 rpm. Generator output is 0.20 v per 1000 rpm, phase shift is 13 deg leading. **72**



STEPPING-SWITCH DIGITAL VOLTMETERS

Ease of servicing was a major design criterion in the new 481A digital voltmeter announced by Non-Linear Systems, Inc., Del Mar, Calif. The four-digit instrument, with ±0.01% accuracy, features plug-in stepping switches.

Use of plug-in switches speeds trouble-shooting and simplifies switch replacement, a potentially troublesome, time-consuming servicing task in stepping switch digital voltmeters. With new "finger-control" leverage bars, stepping switches can be replaced in less than a minute, according to the manufacturer. **73**

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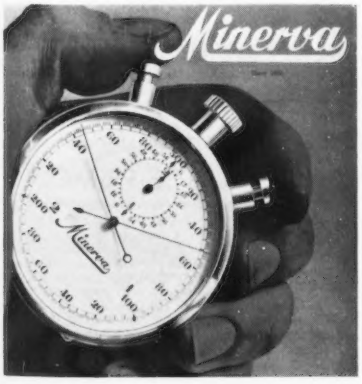
High-temperature Strain Gages

Monograph 26 of the National Bureau of Standards summarizes the work done at NBS in a research program aimed at the development of new types of strain gages for high-temperature use.

Available from the Superintendent of Documents, USGPO, Washington 25, D. C., for twenty cents, the 20-page booklet covers the history of the development of strain gages and the characteristics, development and evaluation of high-temperature properties.



Timing Instruments

A 12-page catalog available from M. Ducommun Co., 580 Fifth Ave., New York 36, N. Y., illustrates nearly 50 different models of stopwatches, chronographs, holders, timers and other timing instruments.

Drawings and descriptions explain exclusive features that determine the quality and reliability of these products **103**

X–Y Recorder

The Model HR-92/93 X–Y recorder for recording one variable in terms of another is described in a two-page illustrated bulletin 792-5 offered by Houston Instrument Corp., P. O. Box 22234, Houston 27, Tex. Featuring flat bed construction, standard $8^1/_2$ x 11-in. paper size and clip-on pens, this plotter is available with a variety of inputs and an optional electric pen lifter. Completely mechanical and electrical isolation provides for no cross-talk between channels and for easy checkout. **105**

Size 8 Components

A 12-page illustrated brochure issued by Kearfott Division, General Precision, Inc., Little Falls, N. J., describes, pictures and tabulates specifications for the complete family of Size 8 components. Presented are: synchros, resolvers, servomotors, servomotor tachometers, synchronous motors, gearheads, brake clutches and permanent magnet alternators. **107**

Analog-computer Handbook

A complete operator's handbook on analog computing is available from Electronic Associates, Inc., Long Branch, N. J., at $1.75 per copy. Written as an operator's manual for EAI's PACE TR-10 transistorized analog computer, the 70-page book provides a comprehensive introduction to the capabilities and operation of general purpose analog computers. The advantages and relative simplicity of analog computers over digital computers in solving many engineering problems is clearly described.

The first part of the three-section book covers the various computing components and their functions, using the components available in the PACE TR-10 as specific examples. Accompanying diagrams and text illustrate the theory and operation of analog computation. The second part introduces the reader to basic programming procedures, leading him through the various steps from the basic system equation to the setting up of the computer to solve it. The third part illustrates the application of analog computation to a series of advanced problems.

Cryogenic Instrumentation

An illustrated data file, CG112R, describing activities in the field of cryogenic instrumentation, is available from Cryogenics, Inc., Stafford, Va.

The two-color heavy stock folder contains information on equipment to accurately measure temperatures, pressure, flow and liquid level for use in cryogenic environments.

Specifications and schematics for the company's line of thermocouples, liquid level detectors and vapor pressure thermometers are included. **111**

Precision Temperature Sensors

Trans-Sonics, Inc., P. O. Box 328, Lexington 73, Mass., has introduced a family of precision temperature sensors for use with its wide range electronic thermometer. As described in its new Special Product Note 140A, surface, bulb and open type platinum-resistance transducers may now be used interchangeably.

The temperature sensors described may be permanently mounted in a vessel, line, duct or on a surface, to be connected, when the need arises, to the wide range electronic thermometer. This equipment reportedly eliminates the need for individual indicators and provides the user with the accuracy and reliability inherent in platinum-resistance thermometers. **113**

Rectangular Meter

A 4 x 6-in. rectangular panel meter line is introduced in a two-color data sheet published by Helipot Division of Beckman Instruments, Inc., 2500 Fullerton Rd., Fullerton, Calif.

The data sheet highlights the classic lines of the 4- x 6-in. meter bezel. It shows photos and outline drawings of the meter, lists standard data, standard specifications as well as modifications available. Back cover of the four-page data sheet lists the 95 ranges available as a-c and d-c voltmeters, ammeters, milliammeters and microammeters. **115**

Ripple Meter

A specification sheet available from Avtron Manufacturing, Inc., 10409 Meech Ave., Cleveland 5, Ohio, describes the uses and features of the model T256 ripple meter as well as listing specification details. **117**

Servomotor Generator

A Size 8 servomotor generator with both motor and generator wound for 115-v, 400-cycle excitation is described in a data sheet published by Helipot Division of Beckman Instruments, Inc., 2500 Harbor Blvd., Fullerton, Calif.

The new Size 8 Model 9008-1106-0 is said to be of special interest to systems designers as a definite aid in simplifying circuitry, achieving greater reliability and cutting costs of new systems.

The data sheet shows photos, electrical specifications for motor and generator, motor dynamics and environmental data. Outline and schematic drawings are included as well as the torque-speed curve. **118**

Complete Product Line

A revised Product Digest No. 160 with pictures and a brief description of their complete product line is available from Midwestern Instruments, Inc., P. O. Box 7509, Tulsa, Okla. The booklet shows Midwestern's standard products and gives general performance information on each one.

The company manufactures both wet-process and direct-readout oscillographs, galvanometers, bridge-balance units and galvanometer amplifiers. They also have a complete line of ground-station telemetry instruments and electro hydraulic servo valves.

Midwestern has built and marketed the Magnecord line of professional tape recorder/reproducers. They also have instrumentation tape equipment and have recently introduced the M-3000 digital tape transport system for use in conjunction with computers and other such applications requiring a high-performance and dependable digital recorder. **119**

Antenna Positioning and Control

A folder giving design and manufacturing capabilities for positioning and control systems of radar and radiotelescope antennas is offered by Datex Corp., 1307 S. Myrtle Ave., Monrovia, Calif. Various systems built by Datex, and now in operation, are described. **121**

Dielectric-constant Analyzer

Describing the many uses to which their D-K Analyzer can be put is a 4-page bulletin offered by Delsen Corp., 719 W. Broadway, Glendale, Calif.

The analyzer, model 168, is a portable, direct-reading instrument for measuring the dielectric constant and the dissipation factor of solids and liquids with the use of a suitable probe. Many uses are suggested for material research and quality control programs. **123**

Size 5 Components

A 4-page technical-data bulletin offered by Kearfott Division, General Precision, Inc., Little Falls, N. J., gives by description, tables and drawings all pertinent information on their family of Size 5 components. The list includes synchros, resolvers, servomotors, motor generators, synchronous motors, gearheads, clutching brakes and brake-clutches. **126**

Product Line Described

The Spring issue of Helinews, published by Helipot Division of Beckman Instruments, Inc., 2500 Fullerton Rd., Fullerton, Calif., is available. This 8-page magazine supplies engineers with illustrated technical information on new a-c potentiometers, delay lines, panel meters, off-the-shelf servomotors and trimming pots. **127**

Circuit Module-amplifier

Specification sheet DPS/A8 covers the CM-114 circuit module-amplifier, a transistorized printed circuit module containing 20 single stage amplifiers for general switch operations. The sheet is available from Datex Corp., 1307 S. Myrtle Ave., Monrovia, Calif. **128**

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Covers the general theory of bending of plates under lateral load, with applications to circular and rectangular plates, and with numerous tables simplifying the calculation of deflections and stresses in plates. Provides a treatment of combined bending and tension or compression of plates, with applications to thin plates. **By S. Timoshenko, Stanford Univ.; and S. E. Woinowsky-Krieger, Université Laval, Quebec, Canada. 2nd Ed., 580 pp., 280 illus. $15.00**

THEORY OF ELASTICITY

For engineers concerned with calculating stresses in engineering structures and machine parts—here is essential knowledge of the theory of elasticity, together with the solution of engineering problems of practical importance. The book discusses approximate and experimental methods of solving elastic problems which have proved useful in the study of complicated cases of stress distribution. **By S. Timoshenko and J. N. Goodier, Stanford Univ. 2nd Ed., 506 pp., 266 illus., $11.50**

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What the bonded wire resistance strain gage can do in solving problems in experimental stress analysis is explained in this practical book. All phases are covered—from selecting the proper commercial gage through preparing surface of the test member, cementing, damage-proofing, and wiring the gage—to selecting the proper instrument to record the strain gage data, and interpreting readings in terms of significant stresses. **By C. C. Perry and H. R. Lissner, Wayne Univ. 280 pp., 180 illus., $6.00**

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Gold-cobalt Thermocouples

A four-page technical bulletin, CG357, describing gold-cobalt vs. copper thermocouples is offered by Cryogenics, Inc., Stafford, Va.

A chart shows comparisons of sensitivities over temperature spans encompassing some normal boiling points. Some possible thermocouple configurations are illustrated along with a typical rake installation, and specifications and available options are listed.

Also described in the bulletin is an indicating thermocouple millivolt- meter for use in production, research, testing and process control, and a vertical scale potentiometer for use with thermocouples. **129**

X-ray Microscope Problems

A new 8-page folder titled "Isolation of Selected Elements with an X-ray Projection Microscope," reprint of an article by a Pomona College physicist, is available from Philips Electronic Instruments, 750 S. Fulton Ave., Mount Vernon, N. Y.

Included in the article are five drawings, six microradiographs and a table giving absorption edge data for 21 selected elements in the atomic scale from No. 6 (carbon) to No. 82 (lead). One curve shows absorption coefficients versus wave length for several elements, another gives density of a film for visible light, and a block diagram illustrates how projection microscopy pictures can be analyzed.

Two diagrams show computations of admissible error in target and film positions, respectively. Author demonstrates how projection X-ray microscopy provides large depth of field, choice of reasonably monochromatic radiation and initial image magnification. **131**

Ultra-thin Tubing

Availability charts listing production limits for ultra-thin welded tubing in two forms—cold drawn and bead reduced—have just been published by Western Pneumatic Tube Co., Kirkland, Wash.

The chart for cold drawn tubing—Data Memorandum No. 60 B—gives the minimum and maximum wall thickness for various tubing diameters from $\frac{1}{2}$ to 4 in. in both standard and special analyses. Wall thickness can be held as thin as 0.004 in. for all sizes of tubing.

Tables of standard and special analyses list types of alloys and also gives military and ASTM specifications to which this type of tubing can be produced.

The chart for bead reduced tubing—Data Memorandum No. 60 C—gives minimum and maximum wall thickness for various tubing diameters from $1\frac{1}{2}$ to 10 in. and lists 86 analyses in which this type tubing is furnished. **133**

Punched-tape Reader

Data Stor Division, Cook Electric Co., 2700 Southport Ave., Chicago 14, Ill., offers bulletin DSD-SR-16 describing Model 84, a new punched-tape reader suitable for "on-line" service in computer, communication and control applications.

The data in the form of coded hole sets are photoelectrically read from prepunched tape and transformed to electrical output pulses. Model 84 is described as a self-contained unit, requiring only the necessary connections to a suitable power source and load. Nominal tape length of 500 (for 3 mil Mylar tape) provides a total punched tape capacity of approximately 60,000 data characteristics. The transport has been engineered to incorporate proved features of the highly reliable military transports developed by Cook Electric Co. for the Atlas, Titan, Minuteman and other missile programs. **139**

General-purpose Analog Computer

A 20-page brochure, AC-6007, describing the model 231R, 100-amplifier, general-purpose analog computer, manufactured by Electronic Associates, Inc., Long Branch, N. J., is available. It describes in detail the basic system, the "building block" design which allows easy expansion and the many automatic features designed for this computer.

Also included is a digest of available readout equipment and a 15- by 19-in., full-color reproduction of the new prepatch panel. **141**

Code Converter, Adder/Subtractor

Specification sheet DPS/A1 gives details of the circuit module, CM-101A, a transistorized printed circuit board designed for code conversion and binary addition or subtraction applications. Offered by Datex Corp., 1307 S. Myrtle Ave., Monrovia, Calif. **143**

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The 2500 Kilowatt A-C Air Arc

Controlled electric-arc discharge techniques have been found to produce the correct aerodynamic environment, of sufficient operating times, required to obtain hypervelocity flight conditions for testing purposes in the laboratory

by N. S. Diaconis and F. C. Foshag

SUMMARY—During the latter part of 1957 and early 1958, consideration was given by the General Electric Co. toward the development of electric-arc discharge techniques to heat relatively large and continuous mass flows of gas to pressure levels and stagnation enthalpy levels comparable to those encountered in atmospheric flight up to satellite velocities. Such a device is required for the study of aero-thermodynamic aero-thermoelastic and heat protection problems associated with hypersonic flight conditions. Testing of small models (approximately 1 in. in diameter) with water-arc and air-arc units (50 to 100 kw dc) is sufficient for materials screening purposes, but larger size models (approximately 6 in. in diameter) must be utilized to obtain adequate structural information.

Parametric studies were conducted to define facility requirements, model sizes and stagnation properties. Two 125,000-kva, 3-phase, 60-cycle a-c flywheel-type generators, located at the Switchgear Development Laboratory of the General Electric Co. in South Philadelphia, were utilized to support this program. These generators have the potential of providing a source energy level of 50,000 kw; however, the initial phase of the program concentrated upon development of a unit capable of providing a jet energy level equivalent to 2500 kw. Such a unit can be adapted for use as a low-density wind tunnel capable of simulating free-flight model stagnation properties to Mach 25.

Nomenclature

Symbols

- d = diameter
- h = enthalpy
- H = simulated flight altitude
- $\dot{m}$ = mass-flow rate
- p = pressure
- $\dot{q}_{ST}$ = model stagnation-point heat-transfer rate
- u = velocity
- E = rate of energy addition to working gas
- R_e = Reynolds number
- M = Mach number
- R = gas constant for air
- R_n = model-nose radius
- S = entropy
- T = absolute temperature

N. S. Diaconis and F. C. Foshag are Specialist, Experimental Aerodynamics, and Project Engineer, respectively, General Electric Co., Philadelphia, Pa.

Paper was presented at 1959 SESA Spring Meeting held in Washington, D. C., on May 20–22.

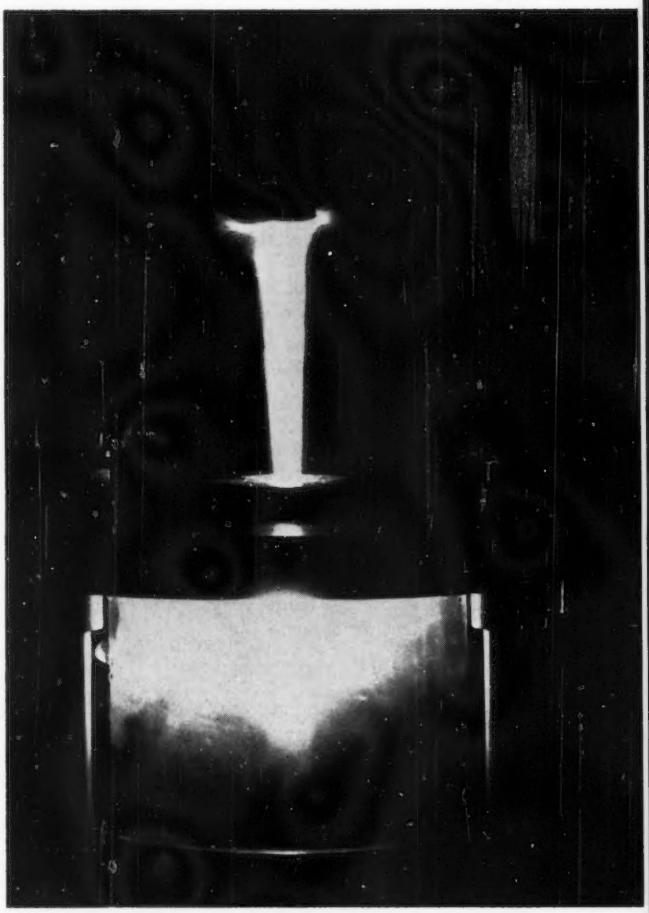


Fig. 1—Water-arc test

Subscripts

- $_0$ = reference conditions, T_0 = 273.16°K
- a = ambient (Pa = 1 atmosphere)
- e = nozzle exit, test section
- t = nozzle throat
- s = stagnation or plenum chamber
- $_{ST}$ = model stagnation-region conditions
- $_\infty$ = free flight

Introduction

With the advent of the ballistic missile and satellite

vehicle, considerable emphasis has been exerted toward the attainment of hypervelocity flight conditions for testing purposes in the laboratory. To investigate the serious materials problems in these various areas properly, it is required that any test facility not only produce the correct aerodynamic environment but be capable of sufficient operating times. In the past few years, it has been found that controlled electric-arc discharge techniques could fulfill these requirements. Large and continuous mass flows of gas can be heated at appreciable pressure levels by electric arc units to stagnation enthalpy values equal to and greater than those encountered in atmospheric flight at velocities up to satellite velocity.

Other present-day facilities having the same high-energy capability have the inherent disadvantages outlined below.

Shock Tubes and Shock Tunnels

Temperature and pressure simulation is possible but test times are available only in the order of milliseconds.

Continuous-flow Wind Tunnels

Test times are almost unlimited, but the stagnation temperatures available are too low.

Rocket Exhaust

The enthalpy level is too low and the chemistry of the flow is complicated and not directly amenable to theoretical analysis.

Aeroballistic Range Testing

Instrumentation problems are quite severe, usually requiring telemetry from very small models, for very short testing times.

Water Arc

The chemistry of the stream is not directly amenable to theoretical analysis and results in a condition that does not simulate flight conditions.

Fortunately, studies to date indicate that the disadvantages of arc facilities—electrode destruction and contamination of the test gas, flow oscillations, pressure limitations on arc-discharging ability—do not seriously restrict their usefulness, for many types of experiments. Developmental investigations now being conducted indicate that, as the state of the art improves, these disadvantages can either be eliminated or their effects reduced considerably.

Advanced Materials Requirements

The successful production of ICBM nose cones depends upon the development of a working heat-protection system capable of absorbing in the vicinity of 2500 Btu per pound. While recent developments in the materials field exhibit a great deal of promise, the fulfillment of this promise in the real re-entry environment, with fabrication processes suitable for full-scale nose cones, still requires the solution of design and fabrication problems.

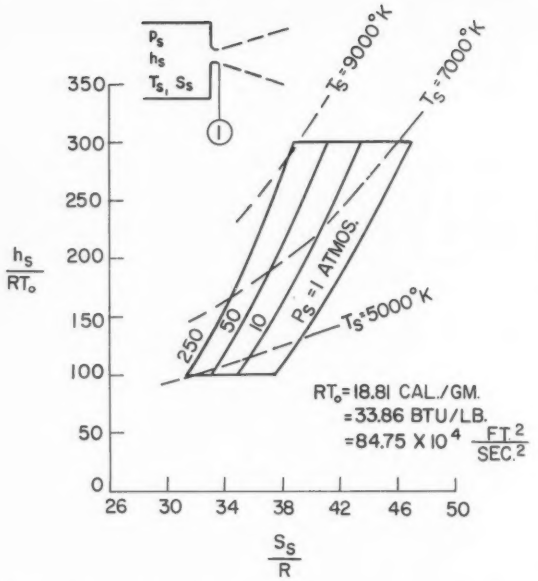


Fig. 2—Range of stagnation conditions considered

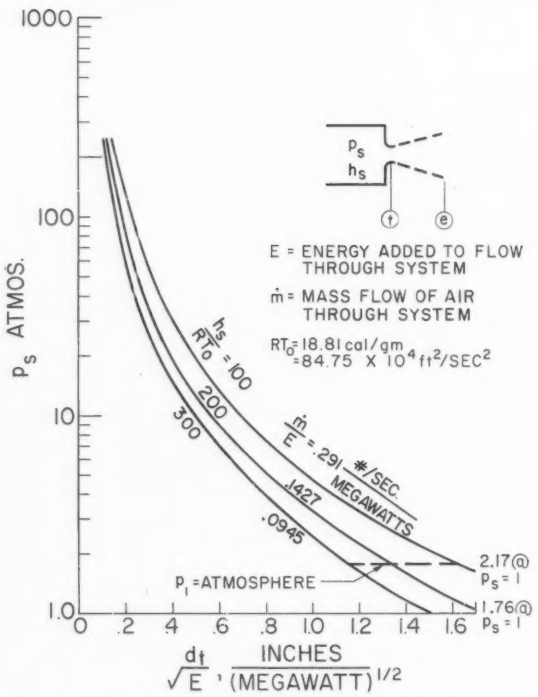


Fig. 3—Throat diameter for high-temperature air jet

Several factors combine to make the design and fabrication problems associated with over-all structural integrity critical: (1) during re-entry the material is in a state of failure near its surface so that

only materials with adequate strength chracteristics are suitable; (2) the mechanical properties of some of the most promising heat protection materials are low; and (3) a minimum thickness of material must be used in order to meet weight requirements. Realistically, severe conditions of ballistic missile re-entry can be simulated by performing atmospheric-discharge free-jet tests; while satellite re-entry studies can be performed by utilizing an electric-arc-heated hypersonic wind tunnel.

The requirements for a ground-based facility for conduction of such tests are as follows:

1. The chemical environment and re-entry thermal gradient into the material must be simulated.
2. Local aerodynamic pressure gradient and shear on test models must be as severe as actual re-entry conditions.
3. Model sizes must be adequate for true simulation of full-scale properties.

A high-power arc facility fulfills these requirements primarily because of its capability for performing large-model tests. Low-power arcs are capable of achieving comparable test conditions, but are inadequate for true evaluation of structural integrity.

General Considerations

Original work with fluid-stabilized d-c arcs was conducted in Germany by Weiss[1] and Maecker,[2] among others. These devices utilized the principle of creating an arc in the presence of a fluid which, thus, became ionized and created a plasma. By restricting this plasma and controlling its release, a high-temperature jet can be created.

This principle was first utilized in our laboratory by constructing a small water-arc unit (Fig. 1). Work was initiated early in 1957 to develop a gas-stabilized unit. While a number of gases can be utilized, our efforts were concentrated on air inasmuch as this was the fluid of interest for re-entry simulation purposes.

Successful development of small d-c air-arc units having a jet-energy level in the vicinity of 50 kw dc served as stepping stones for ultimate development of the 2500-kw a-c air arc. During late 1957 and early 1958, initial design studies and laboratory investigations were conducted under Air Force contract to pursue development of the large unit. A good deal of deliberation resulted over the question of whether d-c or a-c power should be utilized. It was eventually decided that the design should center around the utilization of two large existing 125,000-kva a-c generators inasmuch as they could provide power up to 50,000 kw for a ten-second period. These generators are located at the Switchgear Development Laboratory.

The goal established for the initial phase of development was to provide a unit capable of achieving a jet-energy level equivalent to 2500 kw while the components were designed for a maximum energy

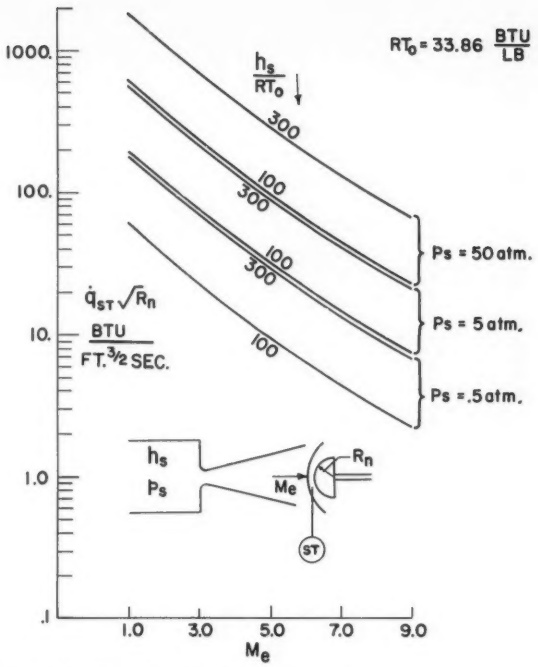


Fig. 4—Stagnation-point heat transfer vs. test-section Mach number

input of 20,000 kw. Advanced stages of development were planned to incorporate a low-density arc tunnel configuration.

All initial designs were predicated on the principle of utilizing graphite for the electrodes and of designing for short-time operation on an uncooled component basis. It was decided that simulated critical gas properties for free-flight Mach numbers to the order of 25 would be utilized for determination of the design area of operation. The area enclosed by the solid line on Fig. 2, over the enthalpy range from an h_s/RT_0 value of 100 to 300, shows the ranges of stagnation conditions that were chosen. It can be seen that an enthalpy ratio of 300 corresponds to a heat content of approximately 10,200 Btu/lb. The maximum operational pressure level of 50 atmospheres was chosen inasmuch as there is not a large practical advantage in terms of flight simulation in operating at a higher pressure level.

Equilibrium calculations based upon the air properties of Hilsenrath and Beckett[3] were made to establish theoretical properties of the air-arc plasma in the above-mentioned area of interest.

The assumptions made were as follows:

(*a*) The flow was in thermal and chemical equilibrium in the stagnation reservoir and during expansion through the nozzle.
(*b*) The flow was one dimensional.
(*c*) The expansion process was isentropic and isoenergetic.
(*d*) Boundary-layer thickness on the nozzle walls was negligible.

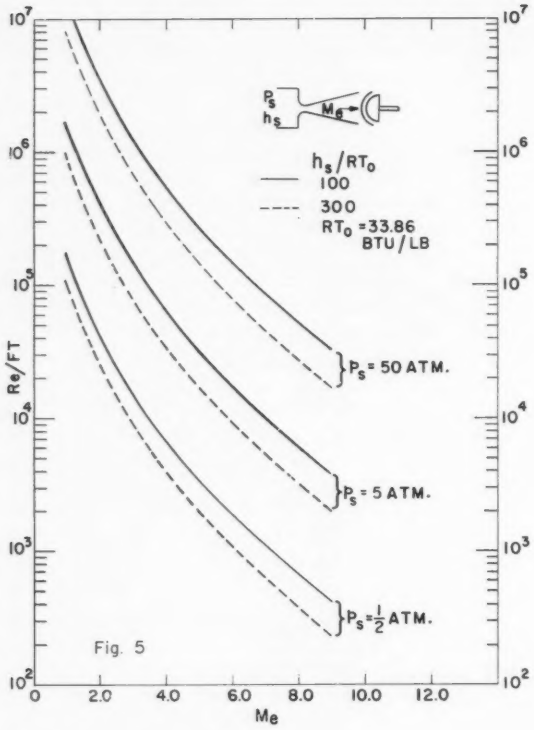


Fig. 5—Test-section Reynolds number vs. test-section Mach number

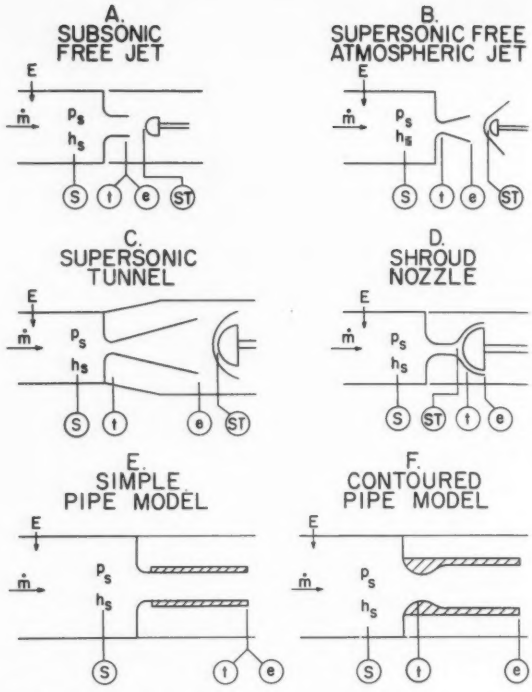


Fig. 6—Test configurations

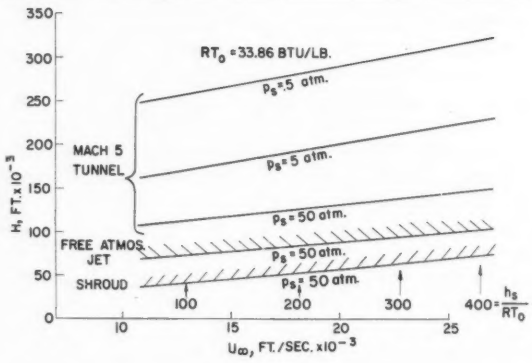


Fig. 7—Stagnation-point property simulation of flight conditions in arc-heated facility

(*e*) There was no effect of any contaminants on the air properties.

No consideration was given to the manner in which energy was to be added to the air upstream of the reservoir or plenum chamber, but the rate at which energy was transferred to the gas E, was left as a free parameter in the results. The variation of nozzle-throat diameter corresponding to the correct throat area for the range of test conditions selected is shown in Fig. 3. As indicated, the air-mass flow through the system is a function of only the energy E and the enthalpy h_s. Values of heat-transfer rates on hemispherical models using the theory of Scala[4] and Reynolds numbers based on model diameter, attainable in arc facilities, are presented in Figs. 4 and 5 for various Mach number flows.

Reproduction of this range of values can be accomplished conveniently by the series of test configurations shown in Fig. 6. These include:

(*a*) Subsonic free jet.
(*b*) Supersonic free-atmospheric jet.
(*c*) Supersonic tunnel.
(*d*) Shroud nozzle.
(*e*) Simple-pipe model.
(*f*) Contoured-pipe model.

Briefly, item (*a*) is a jet in which the test velocity is less than sonic; item (*b*) is a jet where the velocity is greater than sonic at the nozzle exit and the pressure is atmospheric; item (*c*) is similar to (*b*) except that the nozzle exit is usually at pressure levels much less than atmospheric, thus requiring an exhauster system; item (*d*) is a configuration first conceived by A. Ferri,[5] in which the nozzle is so contoured about the model that the local changes in area between the model and nozzle produce the desired test-pressure distribution. To enhance flow steadiness in such a configuration, it is assumed here that a sonic throat or mass-flow choke point is established at some arbitrary location on the model. Item (*e*) is a configuration in which the arc plasma is exhausted from the plenum chamber into a round pipe at the end of which the flow may or may not reach sonic velocity. The test model in this case is the

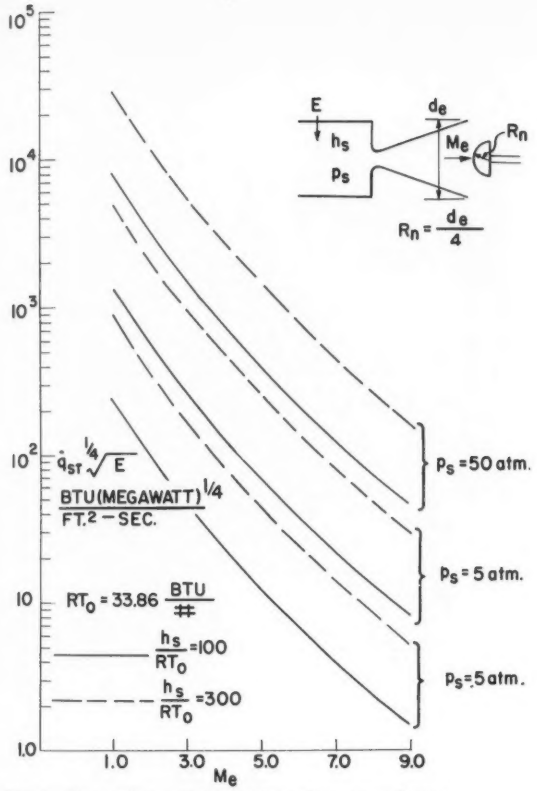


Fig. 8—Stagnation-point heat-transfer power factor vs. test Mach number

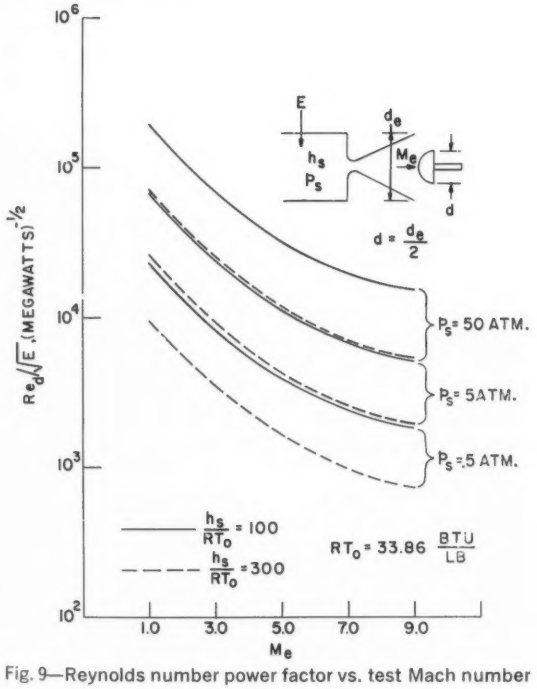


Fig. 9—Reynolds number power factor vs. test Mach number

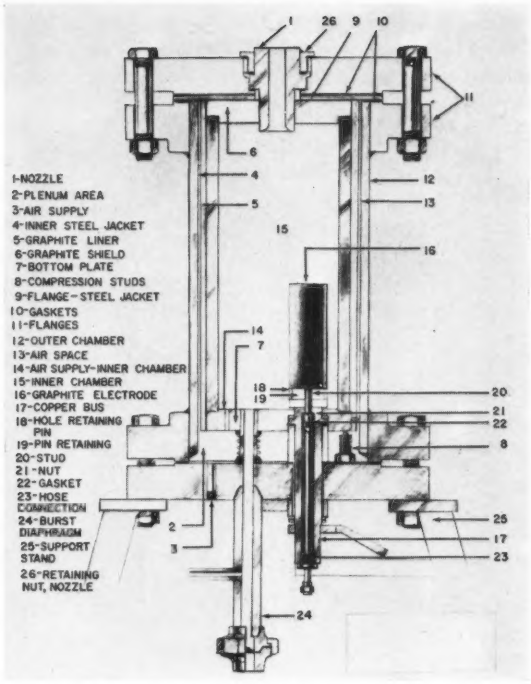


Fig. 10—Sectional view—arc chamber

inside surface of the pipe proper. Item (*f*) is similar to (*e*) except that the pipe walls are contoured to establish a desired test-pressure distribution.

Since stagnation enthalpy is proportional to flight velocity ($h_s \approx u_\infty^2/2$), there is no particular advantage to any of the test configurations considered because heat-addition effectiveness is primarily a function of the arc-heating unit. However, for heat transfer, Reynolds number, flow field and Mach-number simulation, there are considerable differences between the various testing methods. This is illustrated to some extent in Fig. 7 which is a stagnation point simulation plot for the shroud, free-atmospheric jet and a Mach 5 tunnel. The correspondence of stagnation enthalpy, in nondimensional form (h_s/RT_0), and simulated-flight velocity is shown on this plot. As indicated, the shroud and atmospheric jet provide much lower altitude simulation than a high Mach number facility, however, a shroud unit can also provide extensive high-altitude flow properties. Only its lower limit of operation is shown in Fig. 7. The upper limit is determined by the capacity of the vacuum equipment used in a given facility.

The next two figures include considerations of test-section size and facility power. Figure 8 gives the stagnation-point heat-transfer rate, partially normalized by the energy rate E, over a range of test-section Mach numbers. To obtain these results, it was assumed that the model was hemispherical and one-half the test-section diameter. A similar Reynolds number curve is given in Fig. 9. From these results—Fig. 8 and Fig. 9—it is seen that the

Fig. 11—Electrodes, hemispherical

Fig. 12—Electrodes, recessed

test Reynolds number increases with increasing energy input rate whereas the test heat transfer decreases. This is only true of course if the specified relationship between model and test-section size is maintained at each point on the curves.

Operational Development

The present developmental 2500-kw a-c air-arc unit consists of a 1000 psi pressure vessel shown in sectional view in Fig. 10. A burst diaphragm (24) is featured at the bottom to insure against overpressurization resulting from possible nozzle blockage during operation.

Three graphite electrodes (16) of "pie-shaped" design and approximately 10 in. in height are located inside the chamber (12). These electrodes are secured on top of cylindrical water-cooled copper buses (17) which enter through the bottom flange of the chamber. Textolite insulating material is utilized to insulate these buses from the lower flange through which they pass. The electrodes themselves are located immediately above an inner flange of transite (7) which comprises the bottom of the inner chamber (15). The area (2) between the lower flange and the transite flange serves as a plenum area for the air which is introduced into the unit from an adjacent high-pressure source. This air enters the arc chamber proper through six $^3/_8$-in. holes (14) drilled 45 deg from the horizontal to create a rotational flow condition in the chamber.

A graphite liner (5), contained in a steel jacket (4) by means of a shrink fit, surrounds the electrodes and is supported by the transite flange. The transite flange, in turn, is supported by four spring-loaded compression studs (8) to accommodate thermal expansion. A graphite shield protects the upper flange of the steel jacket and features a hole at the center for insertion of the nozzle (1). The nozzle itself is likewise fabricated of graphite and is secured in the upper flange of the assembly.

The air which is introduced into the bottom of the chamber (3) passes upward through the annulus of $^3/_4$ in. between the circular-electrode configuration and the chamber wall. One hole ($^3/_{16}$ in.) through the transite flange at its center point, with $^1/_8$ in. angular holes drilled 45 deg therefrom, introduces air between faces of the electrodes. The air passing upward around the electrodes insures that arcing takes place only in the upper 30% zone of the electrodes where the desired heating occurs. The plenum above the electrodes is provided to allow uniform heating and to provide aerodynamic uniformity prior to introduction into the nozzle. The unit operates at an efficiency up to a maximum of 50%. The remainder of the energy utilized is dissipated within the unit by reason of heat loss to the graphite and steel components.

Introduction of the air at the bottom of the unit serves a threefold purpose: (1) as an operating medium; (2) to sustain arcing in the upper region of the electrodes and (3) as a coolant to the lower portion of the electrodes and the graphite liner of the

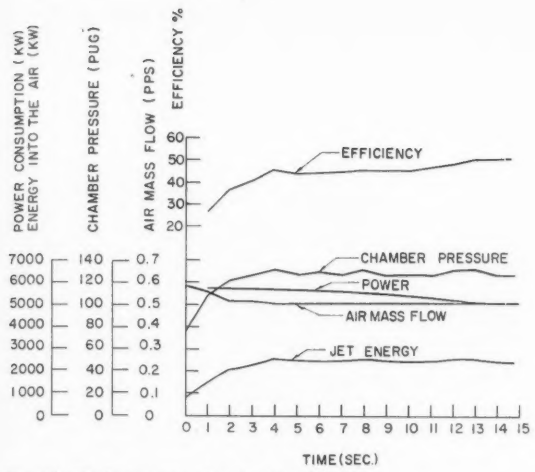


Fig. 13—Performance, development test 5, run 6, May 8, 1958

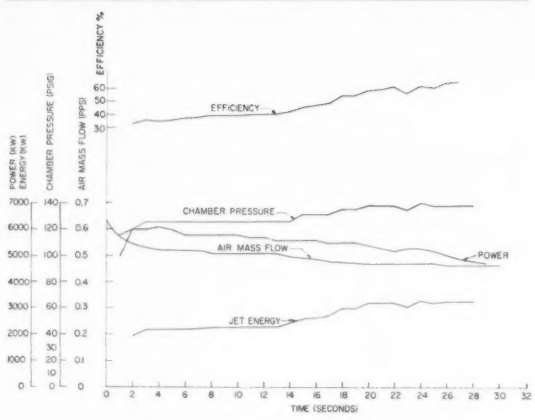


Fig. 14—Performance, development test 5, run 8, May 9, 1958

inner-chamber wall. The maximum temperature which results in the lower portion of the assembly because of this cooling is approximately 400° F.

Three-phase 60-cycle a-c power is delivered by means of copper bus connections to the cylindrical bus connections at the bottom of the unit. The electrical energy passes upward through each electrode and results in arcing phase to phase as well as phase to ground. In the view of the electrode configuration shown by Fig. 11, the conduction pattern when arcing to ground is from each electrode to the graphite wall. The wall in conjunction with the chamber proper is grounded to accomplish this condition.

The upper portions of the "pie-shaped" electrodes are rounded in such a way as to approach a hemispherical shape. This particular configuration (Fig. 11) has been found to function satisfactorily from ambient pressure through 275 psia. Developmental testing revealed that this configuration was unsatisfactory, however, for pressures higher than 275 psia; it was found, instead, that a configuration consisting of an outer ridge with a recessed inner area of the upper portion of the electrode, as shown in Fig. 12, performed satisfactorily up to an operating pressure of 1050 psia. To accomplish this operation, it was necessary to decrease the spacing between the electrodes and the outer chamber wall to $^1/_2$ in. It was likewise necessary to operate at a voltage level twice as great as the level for 275 psia operation. The voltage level for the latter was in the vicinity of 350 v, while it was 700 v for operation at higher pressure. The corresponding current for these conditions was 7000 and 3500 amp, respectively.

Developmental testing has revealed that approximately two ounces of graphite are lost per second of operation. While no conclusive spectroscopic data are available, as yet, it is believed that the great majority of this loss discharges with the air in the form of vapor or carbon monoxide (CO). The ratio of carbon to air resulting from this condition may amount to a maximum of 43% and is dependent upon the operating current level for each particular test. The percentage for normal test conditions is usually about 30%.

The unit described herein features completely uncooled components except for the copper buses which conduct electrical energy to the electrodes. Water is sprayed into the upper portion of these buses, as can be seen by examining the details in Fig. 10, and drains by gravity therefrom. The graphite-lined, steel-jacketed inner liner, which essentially operates with no pressure differential across the wall, is uncooled as is the graphite nozzle. Advanced development tests are in progress, however, which feature cooled components to extend the operating reliability as well as the duration of testing.

To date, the usual duration of a test has been 15 sec. Typical performance obtained is shown by Fig. 13. Test items are introduced at approximately the 5-sec period during a run and are removed at the 15-sec point just prior to disruption of power to the unit. The model is not introduced until the 5-sec point inasmuch as the initial period preceding this point is required for the unit to achieve stable operation. One test of 30-sec duration (Fig. 14) was conducted satisfactorily except that the inner steel-jacketed assembly was overheated. By limiting the operating time to 15 sec, two or three runs are ob-

Fig. 15—Operational view, 2500 kw a-c air arc

tained from each electrode and liner installation. The graphite nozzle is replaced for each test run. Figure 15 represents an operational view of the unit without a test model in the jet.

Aerodynamic-nozzle Development

During the early developmental program (when only mechanical and electrical operating characteristics were being sought), nozzles featuring a 1-in. diam straight bore were utilized. Subsequently, following acceptable mechanical and electrical operation, flow studies were conducted utilizing a conically bored nozzle (Fig. 16). This unit features a straight bored section of approximately 6 in. with the remaining 2 in. bored conically at an included angle of 20 deg. Three different nozzle sizes were fabricated to permit operation at 175, 325 and 765 psia. The throat diameters were respectively 1, 0.656 and 0.40, in. corresponding to flow Mach numbers of approximately 2, $2^1/_2$ and 3. Stagnation enthalpy levels for the tests for which these nozzles were designed correspond to free-flight velocities of 13,000, 18,400 and 22,600 fps, respectively. Fully contoured nozzles have not been utilized to date but the future developmental program includes such work.

Fig. 16—Nozzle, conical discharge, 2500 kw a-c air arc

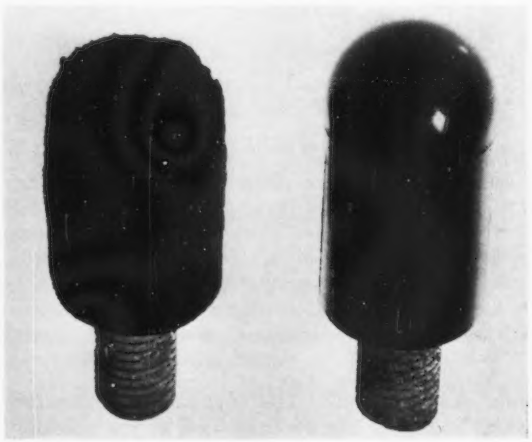
It has been found that, for free-jet test purposes, introducing the model at a distance of $^1/_8$ in. above the nozzle discharge results in the optimum flow conditions being realized. This point is well within the area between the discharge of the nozzle and the first shock diamond. An enlarged view (Fig. 17) of a single 16-mm frame of high-speed film shows a 1-in. diam model during actual test conditions. The ionized gases in the region behind the bow shock wave may be readily seen. The "halo" results from reflection of the bow shock from the jet boundary to the nozzle body and out again to the jet bondary. Figure 18 shows before and after views of a 1-in. diam model of a plastic material tested in this manner.

Fig. 17—Operational view, model test, 2500 kw a-c air arc

Development tests have also been performed utilizing a shroud nozzle configuration with 4-in. diam hemispherical models. The graphite nozzle configuration utilized consisted of a conical shape varying from a diameter of $1^1/_2$ to 4 in. The included angle of this nozzle was 90 deg. Figure 19 shows such an installation prior to testing. Figure 20 presents before and after views of such a model tested in this manner.

Model-test Considerations

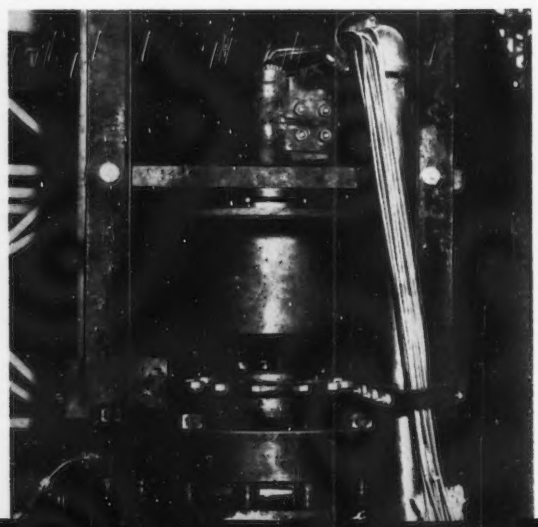
Initial insertion of models into the free jet has been accomplished by means of a pneumatically actuated water-cooled assembly as shown in Fig. 21 (pressure distribution probe in testing position). Figure 22 is a photograph of a $1^1/_2$-in. cone-shaped model prior to testing with this device. The models are moved



After Before

Fig. 18—Test model, 1 in. hemispherical

Fig. 19—Model installation, 4 in. hemispherical



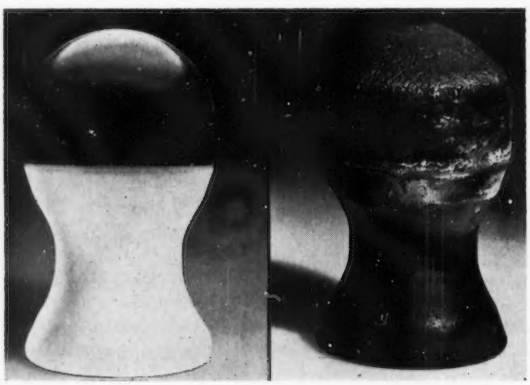
into the jet in a shallow arc in approximately 0.3 sec.

Two alternate designs for model support devices are under development to meet shrouded-nozzle insertion requirements; one will feature a fixed relationship between the nozzle and model prior to isnertion, the other will utilize the cantilevered-arm principle. The former will consist of a complex graphite assembly with the test port moved into position by means of rotary action imparted by a cylindrical actuator. This principle is considered most desirable, providing a satisfactory arrangement can be developed, inasmuch as the nozzle–model relationship can be better controlled. Subsequent plans for this principle include an automatic feed-control system to maintain the nozzle–model relationship constant during a test.

Future Plans

Development plans of the future will be concerned with further exploring operational characteristics of the unit and with achieving a plasma jet with a minimum contamination. The approach being utilized for the latter consists of a completely water-cooled assembly including cooled metal electrodes. Operation has been accomplished to date on a configuration of such electrodes which has resulted in obtaining a jet energy level of 400 kw with an energy input of 2600 kw. This test was conducted for a period of 20 sec. It is to be noted however that the efficiency with a completely cooled configuration is approximately one-half that which was achieved with the graphite configuration. A water-cooled chamber has been designed and fabricated and also has operated successfully for a period of 20 sec in conjunction with the cooled-metal-electrode configuration. A water-cooled nozzle is available but has not been tested to date. It is hoped that ultimate successful operation of these three prime components will result in obtaining a unit which will be capable of test times extending into minutes of duration and which will provide repeated reliable testing without the need for frequent changing of components.

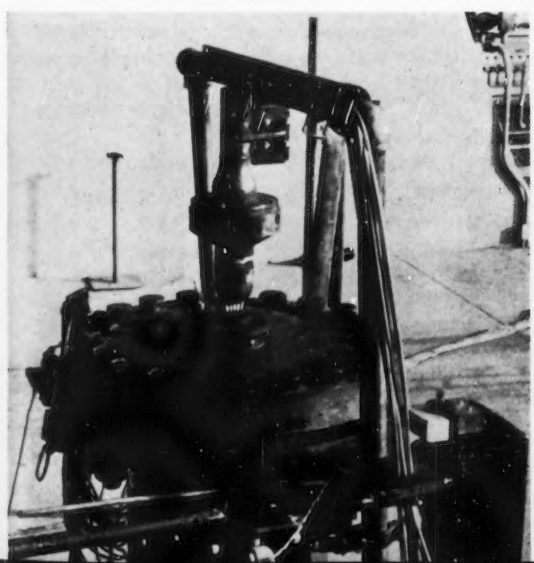
Future designs will consider the use of the arc unit as a wind tunnel component to attain test conditions (long time–low heat flux) associated with satellite re-entry. Investigations of materials with small arc-tunnel facilities have shown that the heat-protection problem in this area differs sufficiently from the ballistic-missile re-entry problem to require large model tests in a tunnel of this nature.



Before After

Fig. 20—Test model, 4 in. hemispherical

Fig. 21—Model holder and actuator



Acknowledgments

The contribution of W. R. Warren who guided and directed the original technical study upon which this work has been based is gratefully acknowledged.

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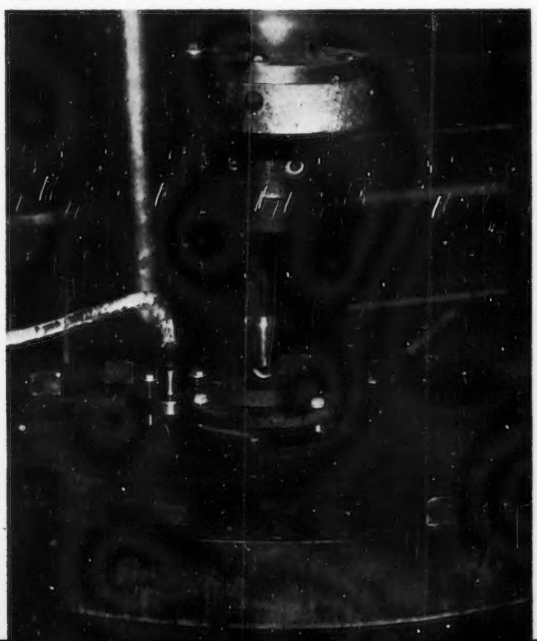
Fig. 22—Test model, $1\frac{1}{2}$ in. cone shaped



Yield Strength of Machined Ring-stiffened Cylindrical Shell under Hydrostatic Pressure

Experimental investigation of the effects of initial imperfections and residual welding and rolling stresses includes tests of a machined and stressed-relieved model identical in geometry and of same material as a previously tested fabricated model

by M. E. Lunchick and J. A. Overby

ABSTRACT—The effects of initial imperfections and residual welding and rolling stresses on the yield strength of a stiffened cylinder were investigated by tests of a machined and stress-relieved model, Model BR-7M, identical in geometry and of the same material as a previously tested fabricated model, Model BR-7. The experimental collapse pressure of 1502 psi agreed well with collapse pressures computed from theories which account for the plastic reserve strength. The pressure at which yielding began agreed more closely with that calculated by the maximum-shear-stress or the Hencky-Von Mises criterion than with that calculated by the maximum-principal-stress criterion. The data also indicated that the mathematical form of the deflection function of the shell did not change appreciably in the elastic-plastic range. Comparison of the collapse pressures of the machined and welded models indicated that, for the geometry tested, the residual welding and rolling stresses do not adversely affect the collapse pressure.

Introduction

To design adequately the shell of a stiffened cylinder for the axisymmetric yield mode, it is first necessary to close the gaps in the knowledge regarding yield failures. For this reason, a research program was initiated at the David Taylor Model Basin to establish the following general objectives:

1. Experimental validation of theory predicting collapse pressure.

2. Experimental validation of the elastic analyses by measurement of strains and deformations within the elastic range.

In the fabrication of models,* geometrical imperfections and residual stresses due to rolling and welding inevitably occur. On the other hand, the theories used to predict deformations and collapse pressure assume an initially perfect and stress-free model. In order to evaluate the theory and to determine the effects of imperfections and residual stresses, a machined and stress-free model was manufactured which was a duplicate of the geometry of Model BR-7, one of the welded models which had been tested earlier.

The machined model, designated BR-7M, is described in this report together with its instrumentation, test and test results. The test results are compared with those from tests of the welded model, and the influence of the imperfections and residual stresses on axisymmetric yielding is evaluated. In addition, the behavior of the model in the elastic-plastic state is examined.

Description of Model

The geometrical properties of the fabricated and machined models are shown in Table 1. A schematic drawing of Model BR-7M is shown in Fig. 1. Essentially, the model is a cylinder stiffened externally by rectangular frames forming eight equal bays. At the ends of the model are two heavy

* *The stiffened cylinders tested will henceforth be referred to as models.*

TABLE I—DIMENSIONS AND YIELD STRENGTHS OF MODELS BR-7 AND BR-7M

Dimension	Model BR-7	Model BR-7M
Thickness of shell, h, in.	0.211	0.211
Mean diameter, D or 2R, in.	26.899	26.899
h/D	0.00784	0.00784
Frame spacing, center to center, L, in.	2.570	2.570
Effective frame spacing, L_f, in.	2.034	2.044
L/D	0.096	0.096
Effective faying width, b, in.	0.536	0.526
Area of frame, A, sq in.	0.404	0.404
Ratio of frame area to shell area, A/Lh	0.745	0.745
Thinness ratio*	0.42	0.46
Compressive yield strength, psi	47,500	59,200

* The thinness ratio is defined as

$$\lambda = \sqrt[4]{\frac{(L_f/D)^2}{(h/D)^3}} \cdot \sqrt{\frac{\sigma_y}{E}}$$

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frames. The two end bays are $^{8}/_{10}$ the length of the other bays. Model BR-7M had 45-deg fillets at the base of the frames of the same size as the welds on the fabricated model; thus, the faying width of the frames was closely duplicated.

The model was manufactured from a $2^{1}/_{2}$-in. flat plate of high-tensile steel (HTS). This plate was rolled at the Norfolk Naval Shipyard to 180 deg of arc. Two pieces of the plate were then welded together with two longitudinal seams diametrically opposite one another to form a thick tube. This thick tube was then machined at the Model Basin to form ring frames integrally with a thin cylindrical shell.

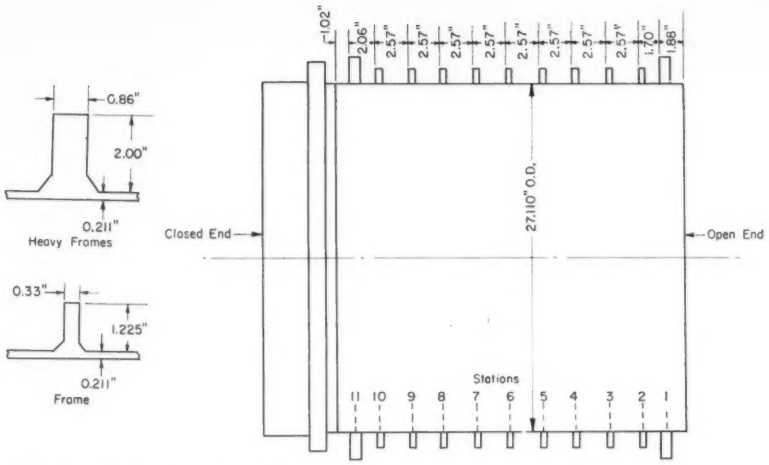

Fig. 1—Schematic diagram of Model BR-7M

The cylinder was stress relieved four times at 1125° F for 3 hr: once after rolling and welding to remove the rolling and welding stresses, twice during the rough machining to remove machining stresses and then after final machining.

The average yield strength of the shell was 59,200 psi. Four compression specimens were taken in the longitudinal direction and four in the transverse direction. Comparing values to the nearest percent, the difference between the average longitudinal and transverse yield strengths was 0%. The difference between the smallest and largest value of yield strength measured was 3%, but most of the values differed from the average by less than 1%.

The out-of-roundness of the model was recorded on the recording deflectometer[1] at all the typical frames (except the one nearest the closure bulkhead) and at the center of all the typical bays. In Table 2, are listed the maximum out-of-roundnesses for Model BR-7M and for the welded Model BR-7. These out-of-roundness measurements are the differences between the maximum and minimum radii of the shell contours from the mean circle. The mean circle was taken as that circle which has its center at the centroid of the circularity contour and an area equal to that of the circularity contour. The circularity charts are shown in Figs. 2 and 3.

Instrumentation and Test Procedure

In planning the instrumentation, two main objectives were kept in mind:

1. The criterion of yielding was to be established.
2. The deformations across a bay were to be obtained in the elastic-plastic range.

To accomplish these objectives, strain gages were located as shown in Fig. 4.

Since failure of the shell usually occurs in the first typical bay at the end of a model, these bays were instrumented with strain gages (see Fig. 4). In addition, a bay in the center of the model was instrumented on the interior and exterior surface at seven locations to obtain an adequate distribution of strains across a bay.

At Station $6^{1}/_{2}$, gages were placed every 10 deg over an arc of 40 deg to detect any formation of buckle lobes in the shell.

Model BR-7M was tested in the 37-in. pressure tank. Pressure was applied in three runs (see Table 3). Strains were recorded at *all pressures* listed in Table 3.

TABLE 2—OUT-OF-ROUNDNESS MEASUREMENTS OF MODELS BR-7 AND BR-7M

Station	Ratio of out-of-roundness to shell thickness, e/h	
	Model BR-7	Model BR-7M
2	0.105	0.057
$2^{1}/_{2}$	0.111	0.071*
3	0.088	0.043
$3^{1}/_{2}$	0.106	0.047
4	0.080	0.047
$4^{1}/_{2}$	0.101	0.043
5	0.100	0.038
$5^{1}/_{2}$	0.107	0.047
6	0.087	0.043
$6^{1}/_{2}$	0.077	0.047
7	0.121*	0.038
$7^{1}/_{2}$	0.093	0.047
8	0.100	0.038
$8^{1}/_{2}$	0.090	0.052
9	0.093	0.047
$9^{1}/_{2}$	0.089†	0.062†
10	0.102	

* Denotes maximum out-of-roundness measured.
† Denotes station in which failure occurred.

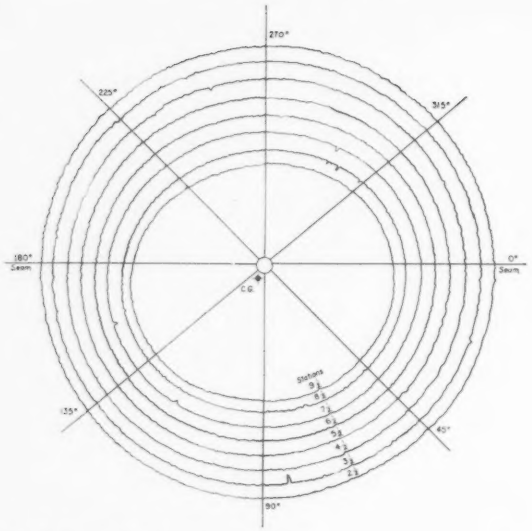

Fig. 2—Initial shell contours

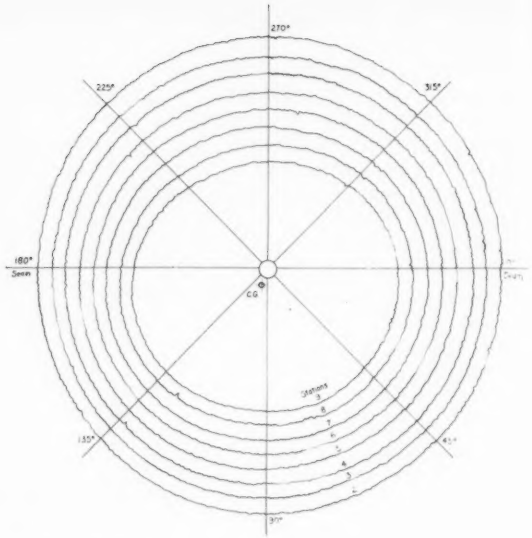

Fig. 3—Initial frame contours

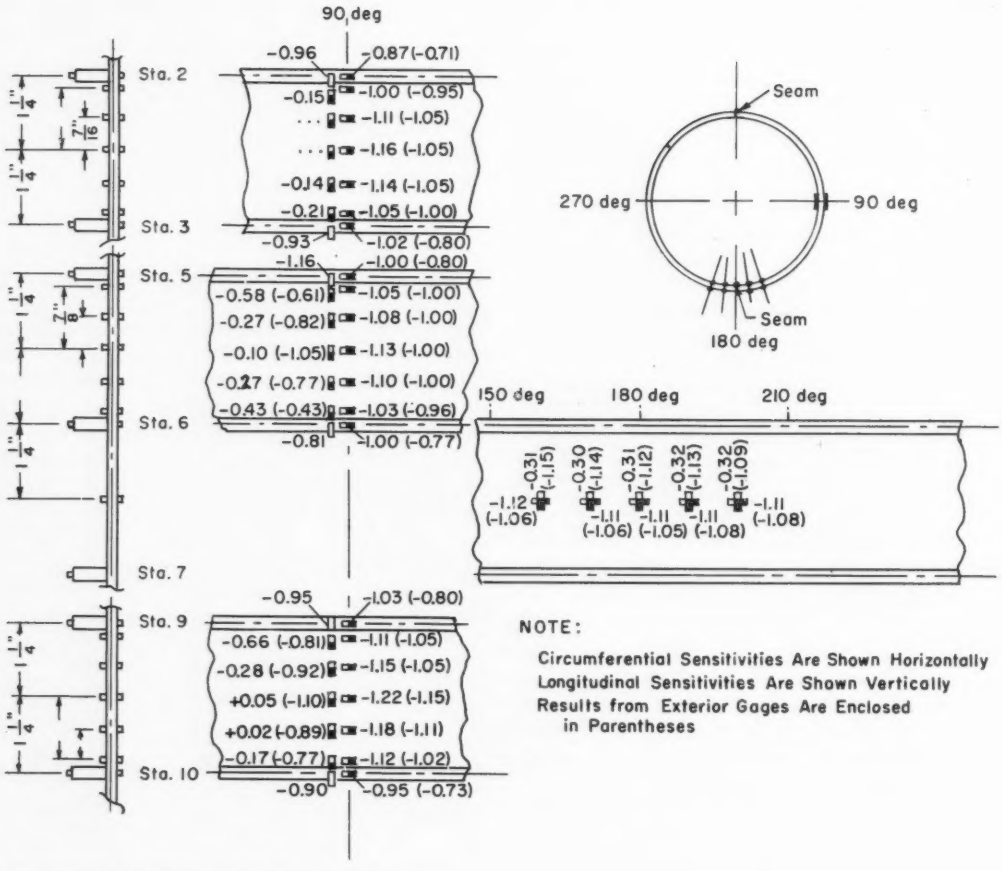

Fig. 4—Strain-gage locations and strain-sensitivity factors

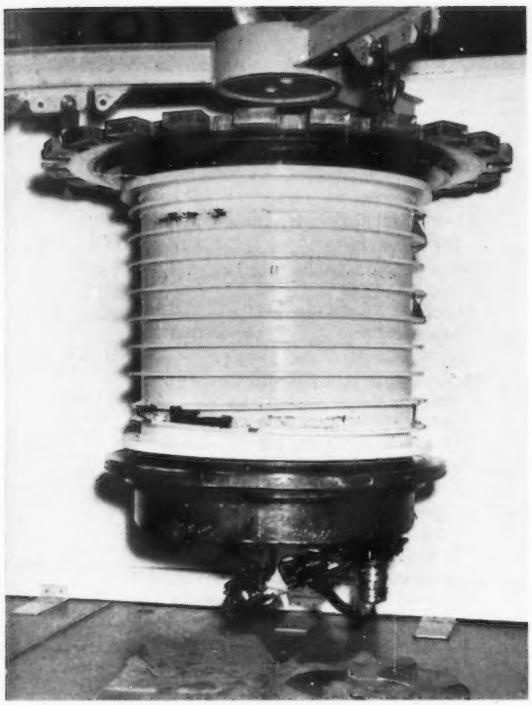

Fig. 5—Model BR-7M after failure

Test Results

Model BR-7M failed at 1502 psi in an axisymmetric yield mode. Damage was visible in the first full-length bays at both ends of the model, Stations $2^1/_2$ and $9^1/_2$, although damage at Station $9^1/_2$ was much greater. Figure 5 shows the model after collapse.

Strain-sensitivity factors were obtained from pressure strain plots. These factors, defined as strains in microinches per psi of applied pressure, are shown in Fig. 4.

Comparison Between Theory and Experiment

Yield Pressure

Since Model BR-7M was machined and stress-relieved, the pressure at which yielding began should not be influenced by residual stresses or geometrical imperfections. Thus, a comparison between this pressure and those computed by various criteria of yielding can be made with validity and should indicate the most appropriate criterion of yielding for the biaxial state of stress encountered in stiffened cylinders under hydrostatic pressure.

The pressure at which yielding began was determined from strains measured at five points on the exterior of the shell at the center of a typical bay. Longitudinal strains were selected because they are more sensitive to yielding than circumferential strains. The longitudinal strains were examined for that pressure at which pronounced nonlinearity

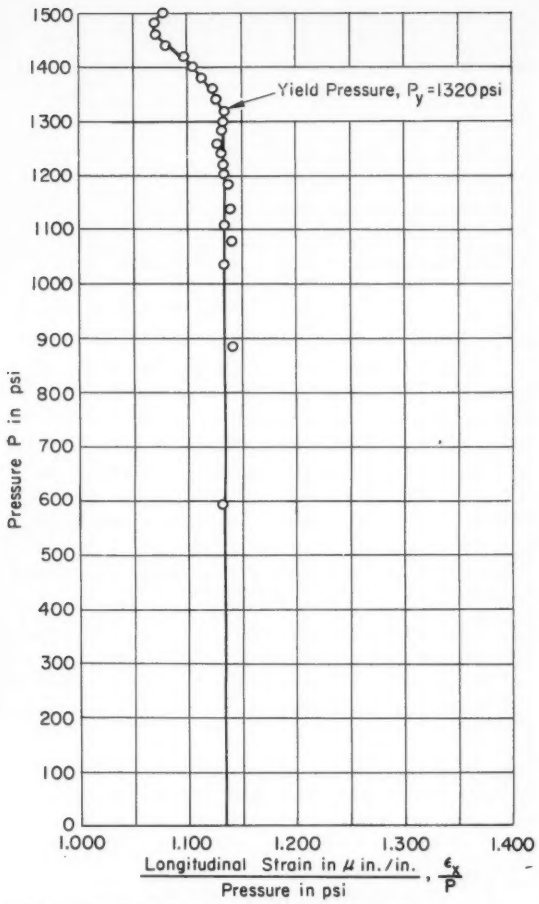

Fig. 6—Plot for determination of yield pressure

TABLE 3—LOADING SCHEDULE FOR MODEL BR-7M

Pressure, psi		
Run 1	Run 2	Run 3
0	0	0
100	200	500
200	400	1000
300	500	1050
400	600	1100
500	700	1130
200	800	1160
0	900	1180
	950	1200
	1000	1220
	1020	1240
	1040	1260
	1060	1280
	1080	1300
	1100	1320
	1120	1340
	1140	1360
	0	1380
		1400
		1420
		1440
		1460
		1480
		1500

TABLE 4—EXPERIMENTAL AND THEORETICAL YIELD PRESSURES FOR MODEL BR-7M

Basis	Yield pressure, psi
Experimental	1320
Maximum-principal-stress criterion	1236
Maximum-shear-stress criterion	1265
Hencky-Von Mises criterion	1259
Maximum-strain-energy criterion	1073
Maximum-strain criterion	1531

in the pressure–strain plots occurred. Essentially, the average of the longitudinal strains at the five points in microinches per inch was divided by the pressure in psi. The pressure at which these quotients began to change markedly was taken as the yield pressure as indicated in Fig. 6. Since the data at five points were used, an average value of the experimental yield pressure could be obtained and is presented in Table 4.

The investigation of the yield pressure considered the following criteria of yielding:[2]

1. Maximum principal stress.
2. Maximum shear stress.
3. Hencky-Von Mises.
4. Maximum strain energy.
5. Maximum strain.

The maximum-principal-stress criteria is especially included as it is currently used in the Von Sanden and Gunther analysis of stiffened cylinders. On the other hand, the maximum-shear-stress criterion is most prevalently used in the analysis of cylinders by the incremental theory of plasticity. The Hencky-Von Mises criterion has strong support by prior experimental and theoretical results. The last two criteria, maximum strain energy and maximum strain, have long been considered obsolete. All criteria, however, are considered for the sake of completeness. Yield pressures computed from these criteria and the measured strains are listed in Table 4.

From Table 4 it can be observed that the maximum-shear-stress and the Hencky-Von Mises yield criteria agree best with the experimental yield pressure. The maximum-shear-stress criterion gives a value 4.2% lower than the experimental pressure, while the Hencky-Von Mises criterion gives a value 4.6% lower. On the other hand, the maximum principal-stress criterion used in Formula [92] of Von Sanden and Gunther[3] is 6.4% low, and the maximum-strain and maximum-strain-energy criteria are definitely not valid for predicting yielding in the model tested.

Collapse Pressure

In Table 5 are listed the pressures calculated by a number of theories which are based on collapse by axisymmetric yielding. In the calculations, the yield strengths were taken as 47,500 psi for Model BR-7 and 59,200 psi for BR-7M. The modulus of elasticity used was 30,000,000 psi. Poisson's ratio

TABLE 5—COMPARISON OF THEORETICAL AND EXPERIMENTAL COLLAPSE PRESSURES

Line		Model BR-7	Model BR-7M
1	Experimental collapse pressure, psi	1300	1502
2	Theory of Von Sanden and Gunther,[3] Formula [92]		
	Theoretical pressure, psi	997	1243
	Theoretical pressure/Experimental pressure	0.767	0.828
3	Theory of Von Sanden and Gunther, Formula [92a]		
	Theoretical pressure, psi	1054	1314
	Theoretical pressure/Experimental pressure	0.811	0.875
4	Theory of Von Sanden and Gunther, maximum-principal-stress criterion at midplane and midbay		
	Theoretical pressure, psi	1112	1386
	Theoretical pressure/Experimental pressure	0.855	0.923
5	Theory of Von Sanden and Gunther, [92a], with Hencky-Von Mises criterion at exterior midbay point		
	Theoretical pressure, psi	1114	1388
	Theoretical pressure/Experimental pressure	0.857	0.924
6	Theory of Salerno and Pulos[5] with Hencky-Von Mises criterion at exterior midbay point		
	Theoretical pressure, psi	1119	1394
	Theoretical pressure/Experimental pressure	0.861	0.928
7	Theory of Von Sanden and Gunther with Hencky-Von Mises criterion at midplane and midbay		
	Theoretical pressure, psi	1235	1539
	Theoretical pressure/Experimental pressure	0.950	1.025
8	Theory of Lunchick[4] predicting fully plastic hinge at midbay		
	Theoretical pressure, psi	1210	1510
	Theoretical pressure/Experimental pressure	0.931	1.005

was assumed to be 0.3. To facilitate comparison of experiment with theory, ratios of theoretical to experimental collapse pressures are given.

The Hencky-Von Mises yield criterion gives pressures closer to the experimental pressures than does the maximum-principal-stress criterion. Von Sanden and Gunther Formulas [92] and [92a] based on a maximum-principal-stress criterion give pressures 12 to 24% lower than the experimental collapse pressure. On the other hand, the pressures based on the Hencky-Von Mises criterion and Formula [92a] are only 8 to 14% lower than the experimental collapse pressure.

The best agreement between experiment and theory is obtained when allowance is made for the plastic reserve strength of the cylinder between the initiation of yielding and final collapse. This allowance can be made simply, but arbitrarily, by neglecting bending stresses and using only membrane stresses to predict yielding (see lines 4 and 7 of Table 5). Again, the use of the Hencky-Von Mises

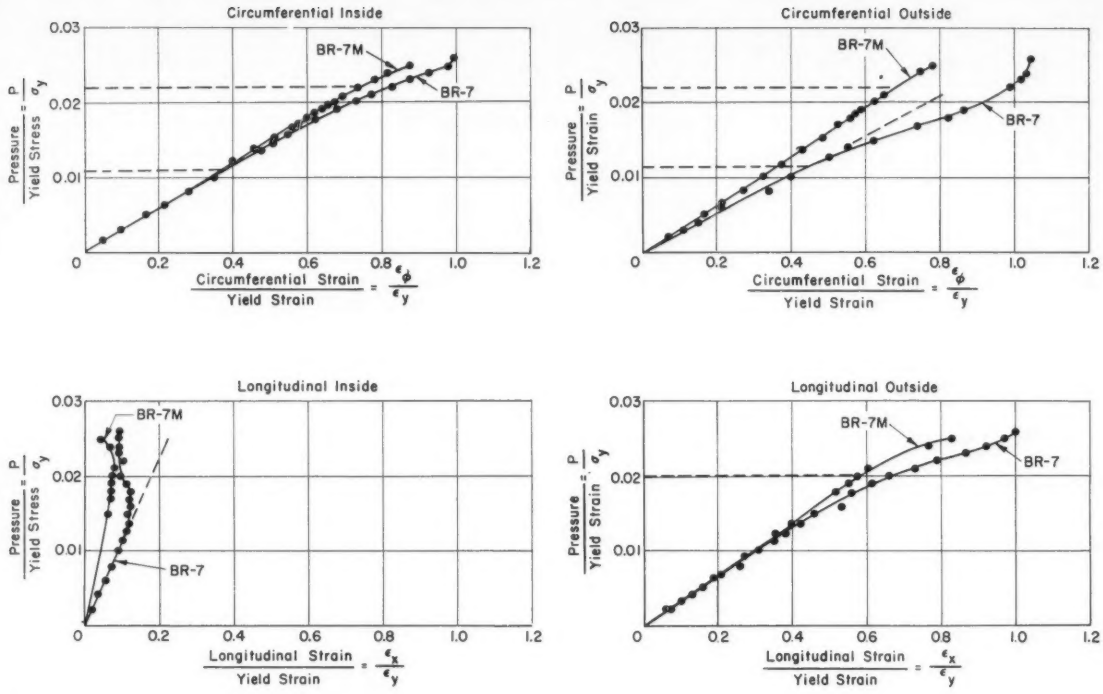

Fig. 7—Comparison of pressure-strain plots between welded and machined models

yield criterion of line 7 in Table 5 gives better values than the maximum-principal-stress criterion. The use of the Hencky-Von Mises criterion at the midplane and midbay gives pressures within 2 to 5% of the experimental pressures. However, a more exact method developed by Lunchick[4] which involves an elastic-plastic analysis of the shell predicts the collapse pressure more closely for machined models.

Since the machined model is closer to the idealized cylinder assumed in the theory than the fabricated model, its collapse pressure would be expected to agree better with that calculated from theory; see Table 5.

Comparison of Welded and Machined Models

Yield Pressure

Yielding in a fabricated model begins at a pressure approximately one-half the pressure at which it begins in a stress-free machined model (see Fig. 7). This observation had been made earlier in many other tests of models. Curves are nondimensional to eliminate any difference in yield strength between the two models. In addition, the curves are for strains at the center of a typical bay, where yielding is considered critical. The differences in pressure at which nonlinearity is pronounced can be attributed to residual stresses induced by rolling and welding. These stresses are known to be high, some exceeding one-half the yield stress,[6] and appreciably reduce the pressure at which yielding begins.

The question may arise concerning the role of imperfections in reducing the yield pressure. Imperfections can be of two types, axisymmetric and asymmetric (commonly referred to as out-of-roundness). For external framing, which is the case in Model BR-7M, the initial axisymmetric deflection actually delays, rather than encourages, yielding at midbay. Any yielding of the shell by out-of-roundness would not initiate completely around the circumference of the shell but only at the troughs of the out-of-roundness contour. At the pressure at which yielding occurs for Model BR-7M, the stresses directly attributable to imperfections are much smaller than the residual rolling and welding stresses, which would, therefore, play the dominant role in reducing the yield pressure.

Collapse Pressure

Although the residual stresses lower the yield pressure, they did not adversely affect the collapse pressure of Model BR-7. However, previous test results[7] showed that welded models failed at lower pressures than machined models when failure was in the shell-buckling mode.

The strength of the machined and welded models can be compared in Table 5. This table lists nondimensional ratios of theoretical to experimental collapse pressure for the two models, thus eliminating any influence due to differences in yield strength between models.

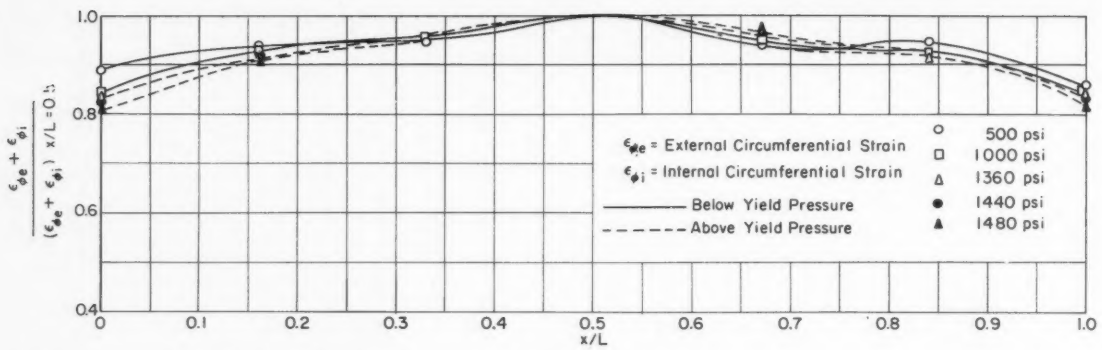

Fig. 8—Deflections of shell in elastic and elastic-plastic states

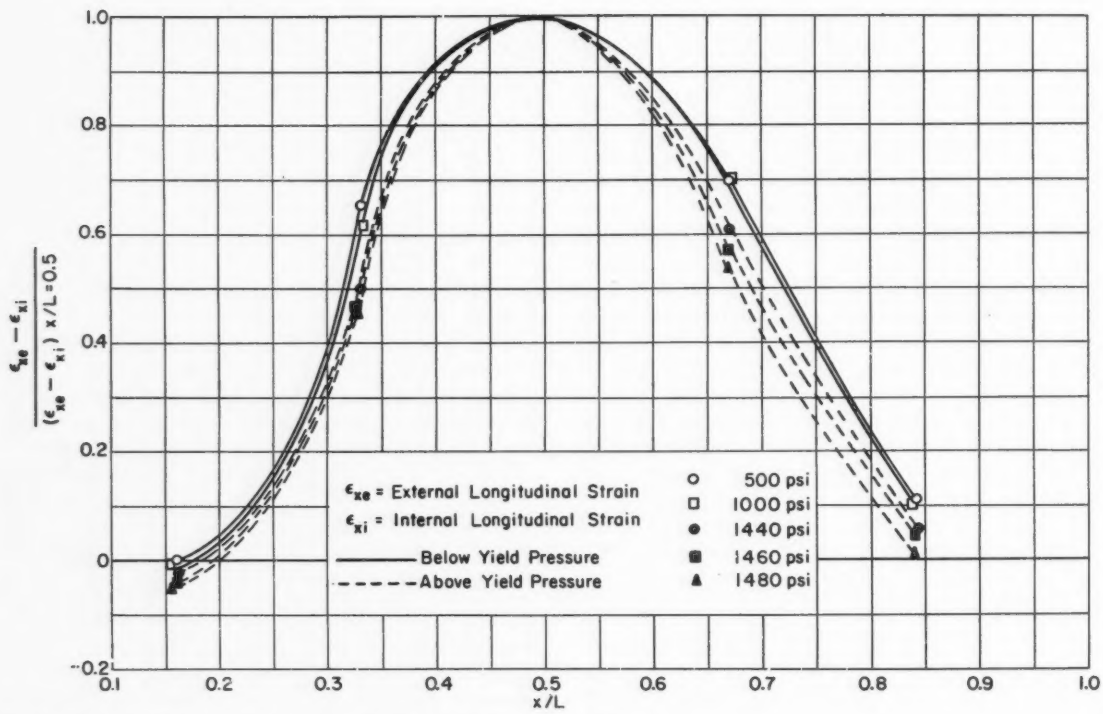

Fig. 9—Changes of curvature of shell in elastic and elastic-plastic states

If the experimental collapse pressure for Model BR-7 is multiplied by the ratio of yield strength of Model BR-7M to that of Model BR-7, the "adjusted collapse pressure" for Model BR-7 becomes 1620 psi, about 8% higher than Model BR-7M. This indicates that, for the same yield strength, the welded model was actually stronger than the machined model. This can be explained on the basis that the welded model had an initial axisymmetric deflection that was beneficial as it had external frames.[8] Also, the collapse pressure of the fabricated model was probably not influenced by out-of-roundness as it was very small, about $^1/_{10}$ the shell thickness, and was not in the same mode as the axisymmetric yield failure.

Deflection Function Before and After Yielding

In addition to investigating the strength of a stiffened cylinder, the test of Model BR-7M was designed to provide information on the strains and deflections after yielding begins. This information would prove useful in the development of theoretical

expressions for the stresses and strains in the elastic-plastic state.

One assumption which could be made in developing a theory is that the radial deflections are symmetrical with respect to a plane midway between ring stiffeners. To meet this assumption, the strains should be studied in a bay at midlength on the cylinder to avoid any influences from the ends. Thus the data presented are for a typical bay between frames 5 and 6.

The deflection function can be examined from Fig. 8 for pressures above and below that at which yielding begins. In Fig. 8, the ratio of the average internal and external circumferential strains at a point to that at midbay is plotted against distance across the bay. (Circumferential strain is indicative of deflection which is the product of this strain and the radius to the point.) By plotting the ordinate in Fig. 8 as a ratio of strains, a single curve would theoretically be obtained if the deflections are elastic and linear with pressure. Figure 8 indicates that this, in general, occurred as the curves for pressures below the yield pressure almost coincide, although they deviate somewhat from one another at one frame ($x/L = 0$). Moreover, Fig. 8 indicates that the form of the deflection function does not change appreciably after yielding begins (see curves for pressures higher than the yield pressure).

The changes in curvature of the shell before and after yielding were also examined. The change of curvature, a function of the second derivative of the deflection for elastic deformations, should be more sensitive to change in shape of the deflected surface than the deflections. The differences in the longitudinal strains on the internal and external surfaces are indicative of change of curvature, since these differences represent the rotation of a line element through the thickness of the shell. In Fig. 9, ratios of the differences of the internal and external longitudinal strains at a point to those at midbay are plotted across the bay. For pressures at which the deformations are elastic and linear, the curves of these ratios should theoretically coincide. As seen in Fig. 9, the curves for pressures below the yield pressure were practically coincident. For the pressures above the onset of yielding, the curves show a greater departure from the elastic curves than Fig. 8. In general, however, change of curvature plots for the elastic-plastic deformations agree fairly well with those of the elastic deformations.

Conclusions

1. The criterion of yielding for the machined model is either the maximum-shear-stress or the Hencky-Von Mises yield criterion. Further tests would have to be conducted before the proper yield criterion can be definitely established.

2. The residual and rolling stresses induced in a welded, stiffened cylinder reduce the pressure at which yielding begins to approximately one-half that in a stress-free machined cylinder.

3. The collapse pressures for the machined model is most accurately predicted by the elastic-plastic analysis of Lunchick given in Reference 4. The elastic theory of Von Sanden and Gunther, coupled with the Hencky-Von Mises yield criterion applied at midplane and at midbay, also closely predicts the collapse pressure.

4. The strength of a welded stiffened cylinder failing by yielding can be as great as one formed by machining if the frames are external.

5. For the geometry tested, the deflection function after yielding begins remains approximately the same as that prior to yielding at pressure close to the collapse pressure. Hence, if theoretical expressions can be developed either for equilibrium or for the strain energy in the elastic-plastic state, the deflection function can be used together with one of these expressions to determine the relationship between pressure and strains.

Acknowledgments

The authors are grateful for the arrangements made by A. R. Willner of the David Taylor Model Basin for rolling the cylinder, examining the model for defects, and stress-relieving during the various states of manufacture. M. A. Krenzke assisted in the instrumentation of the model and the conduct of the test.

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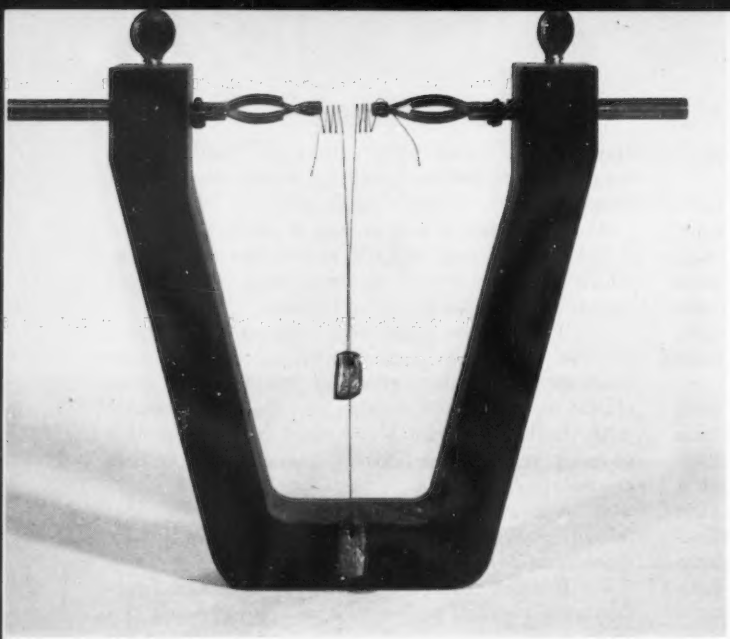

Fig. 1—Jig used for winding electrical-resistance strain elements

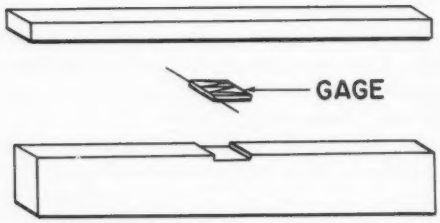

Fig. 2—Method of construction of calibration beam

Use of Electrical-resistance Strain Elements in Three-dimensional Stress Analysis

Strains at interior points of solids can be evaluated through use of a technique that permits the imbedding of strain gages in a plastic model without materially altering the stress pattern in the model

by R. I. Brasier and R. C. Dove

ABSTRACT—A method for constructing resistance-wire strain gages which may be imbedded in a plastic model without materially altering the stress pattern in the model is presented. The methods used to calibrate the gages and the photoelastic tests made to investigate the effect of the gages on the stress pattern are described. The application of this new three-dimensional technique to evaluate the stress distribution in a thick-walled cylindrical pressure vessel is discussed. Correlation between experimental data and calculated values is given.

Introduction

Three-dimensional experimental stress analysis is possible using three-dimensional photoelastic techniques, but this method is complicated by (1) the required optical properties of the model material, (2) the necessary thermal treatment of the model during loading, (3) preparation of the model for analysis (as by slicing) and (4) the analysis of the photoelastic fringe pattern to deduce stresses.

At the present time, electrical-resistance strain gages are used primarily for evaluating surface strains. The investigation discussed in this paper was undertaken to extend the use of electrical-resistance strain gages to the evaluation of strains at interior points of solids. This method required a model of the device or machine member to be analyzed. However, the model may be loaded as many times and in as many different ways as desired, since the stresses need not be frozen in. For any load condition, the strain at the gage point is measured directly using the usual strain indicator. The original model may be modified, and the effect of the modification determined.

Gage Development

One mil Advance wire was selected as the most suitable resistance element because of its relatively high gage factor and low-temperature coefficient of resistance. Of the two standard gage types,

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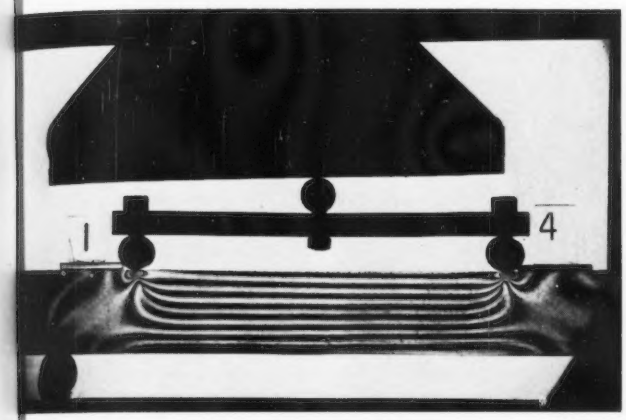
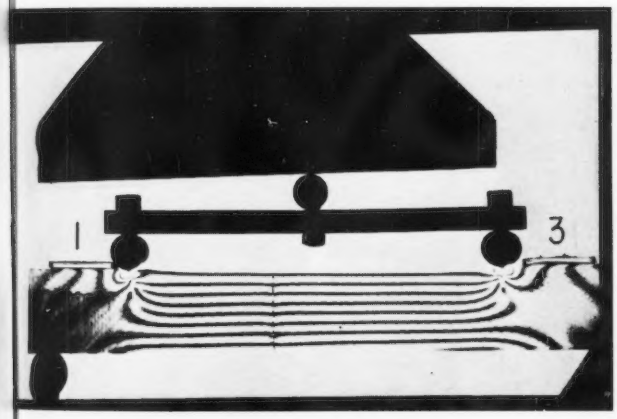
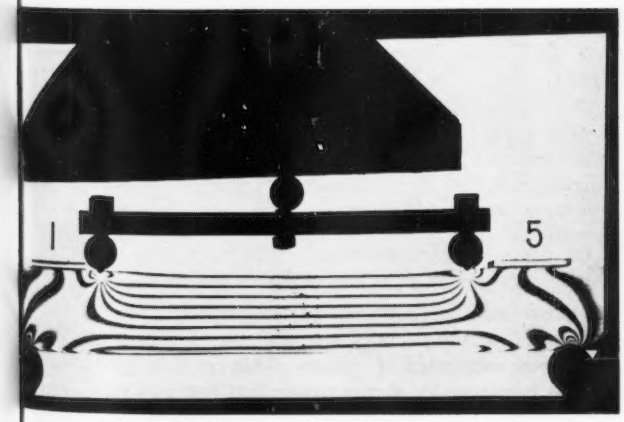
flat grid and wrap-around, the wrap-around type proved to be the easiest to make and include within a model. The jig used to wind the gages is shown in Fig. 1. (For purposes of the photograph, the wire was replaced by ordinary black thread.) The helical grid was wound on a wafer (0.010 in. thick)



(a) Plane of gage parallel to edge of beam



(b) Plane of gage parallel to end of beam



(c) Plane of gage parallel to face of beam

Fig. 3—Strain element imbedded in photoelastic model

TABLE 1—UNIAXIAL GAGE FACTORS

Gage no.	Gage length, in.	Gage factor, compression	Gage factor, tension
1	1/4	1.41	1.50
2	1/4	1.42	1.43
3	1/2	1.46	1.45
4	1/2	1.43	1.41

of the material used for the model. Because of its good photoelastic and mechanical properties, Castolite* was chosen as the model material. The wire grid was cemented to the wafer with additional Castolite plastic. The gage was then ready to be located as desired in the model.

The model was constructed of Castolite mixed in the proportion of 120 g of resin to 6 drops of cumine hydroperoxide. The as-cast material was allowed to cure for 24 hr at room temperature and was then annealed from 200° F. Residual stresses, edge effects and mechanical properties of the material were found to be dependent on this annealing cycle.

The method used for inserting these gages in the model is shown in Fig. 2. In this case the model is a rectangular bar, which was used both for checking the effect of the gage inclusion on strain distribution and for calibrating the gage element. The edges of the model to be joined are cemented together with additional Castolite.[1-3]

The bars containing gages were loaded as simply supported beams with a constant moment section and analyzed photoelastically to determine if the presence of the gage affected the strain distribution. Photographs of the isochromatic patterns are shown in Fig. 3. In Fig. 3a, the isochromatic corresponding to the fringe 2.5 passes through the imbedded gage without noticeable deviation. Hence, the state of strain at the gage location is the same as that which would exist if the gage were not present. In Fig. 3b, a slight disturbance of the strain pattern at the edge of the gage coupon is noticeable. In this case the gage element, which is parallel to end of the beam, extends over almost the full depth of the beam, and over the full depth the strain pattern at the gage location is essentially the same as at any other transverse location. Inclusion of the gage parallel to the face of the beam appears to have no effect on the strain distribution near the gage location (see Fig. 3c). The dots seen in Fig. 3c are small air bubbles trapped when the gage was put in place.

The bars containing gages were tested as shown in Fig. 4 to determine the uniaxial gage factor for two sizes of gages. The results are shown in Table 1.

These same bars were compressed in a direction perpendicular to the gage axis in order to determine the cross sensitivity of the gages.[4] The cross sensitivity (K) was found to be -12.5%.

* *Castolite is the trade name of a polyester casting resin made by the Castolite Co., Woodstock, Ill.*

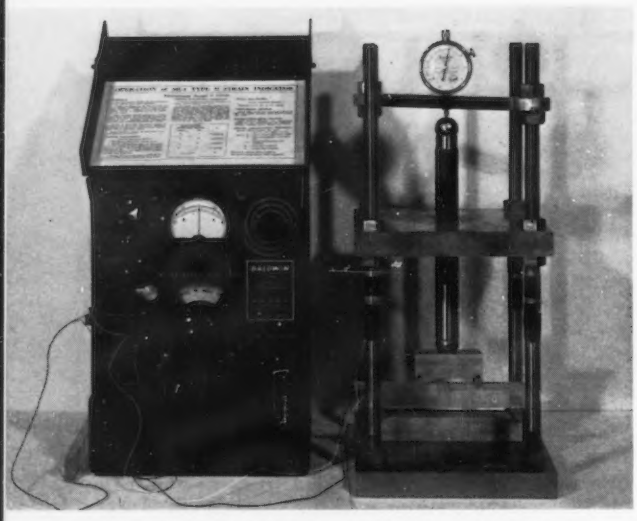


Fig. 4—Experimental determination of gage factor by means of calibration beam

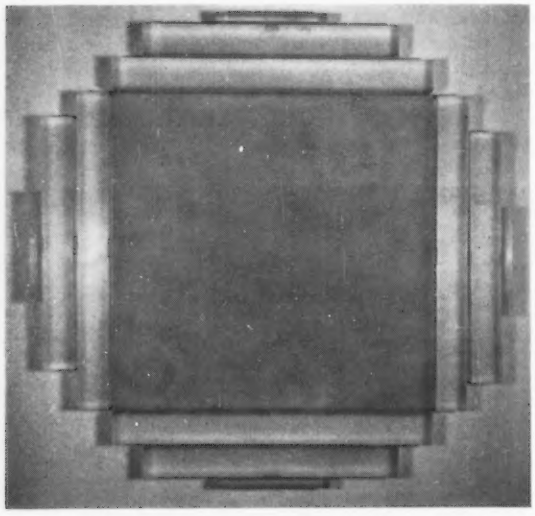


Fig. 6—Center section of pressure vessel before assembly

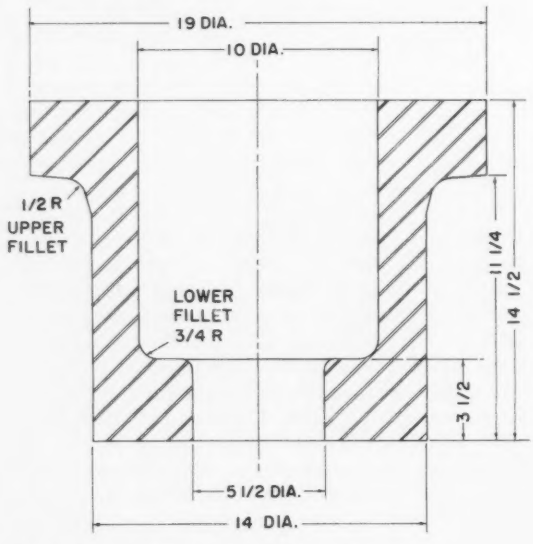


Fig. 5—Cross section of pressure-vessel model

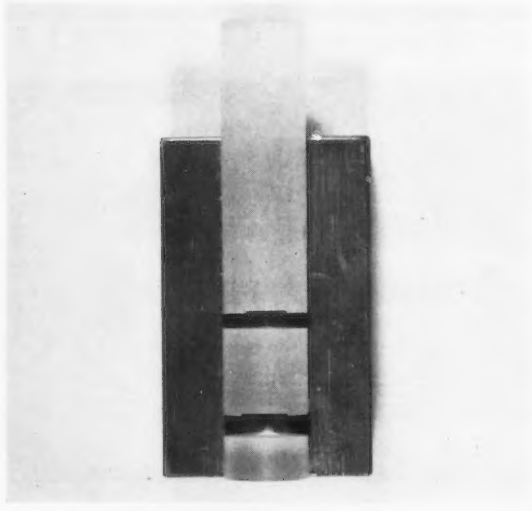


Fig. 7—Castolite plug before assembly

Experimental Procedure

Construction of Model

A cross section of the model is shown in Fig. 5. This model was a $^1/_4$-scale Castolite-plastic model of the outer jacket of a proposed pressure vessel. The model was constructed of sheets of plastic cemented together. The laminations were approximately 1 in. in thickness. Larger castings could not be poured because of undesirable exotherm which leads to highly stressed castings.

Two techniques were used to imbed the gages in the model. One method was to make up sections of the vessel from strips of Castolite which had the gage elements cemented between them. This method (which was the same one used for the calibration beams) was used to construct the center section of the pressure-vessel model. Figure 6 is a photograph of a typical array of strips before they were cemented together.

In the second method, the model section was made up with no gages in it. Then a hole was drilled through the model and a Castolite plug containing the gages was cemented in place. Figure 7 is a photograph of the parts of one such plug before the gages were cemented in place. This method was used to locate gages at the upper and lower fillets. The completed model contained 40 strain gages. Figure 8 is a half-section through the gages at one of four equally spaced locations. Each section con-

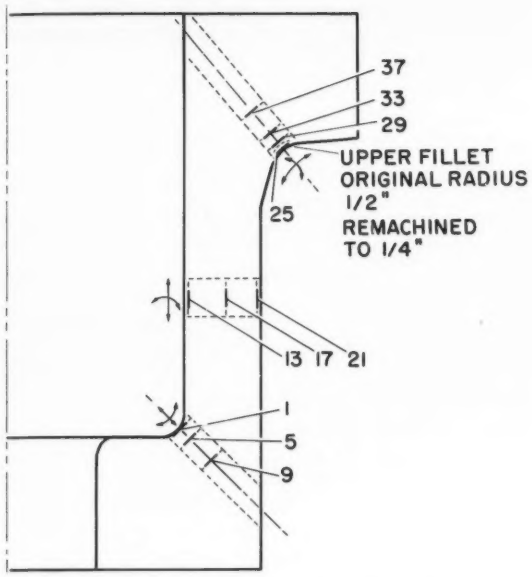


Fig. 8—Section showing method of construction and location of gages

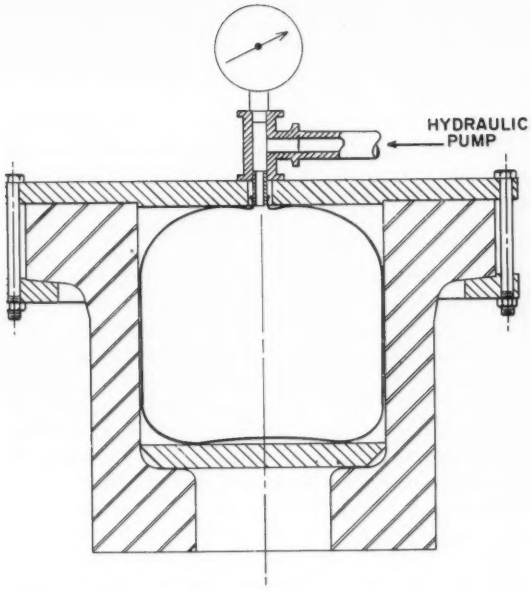


Fig. 9—Pressure test of thick-walled vessel

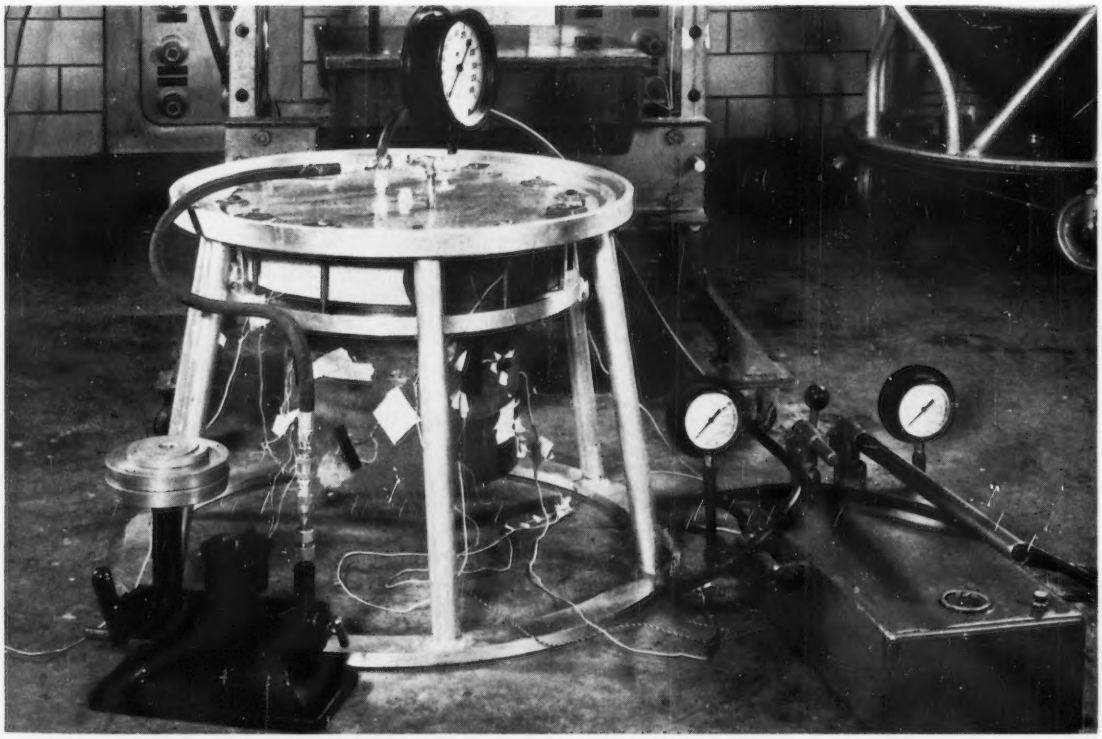


Fig. 10—Apparatus for internal pressure test

tained 10 gages as shown. Gages were included at various radii in the center section so that the experimental results could be checked using thick-walled pressure-vessel theory. Since the gages have a cross sensitivity, both tangential and longitudinal gages were required to reduce the gage readings to strain values.

Two sizes of gages were used. In locations where a high stress gradient was expected, gages with a $^1/_4$-in. gage length were used; where a lower gradient

was expected, gages with a $^1/_2$-in. gage length were used. It should be noted that, even with the smaller gages, the output represents the average strain in a volume $^1/_4$ x $^3/_8$ x 0.012 in.

The lead wires used were 0.004 in. in diameter. They were brought out through the wall of the model and connected to larger wires which were, in turn, connected to the strain indicator.

Test Methods and Results

The model was first tested for the effects of internal pressure. Top and bottom plates were added as shown in Fig. 9 and internal pressure was applied by filling the rubber bag (inner tube) with hydraulic fluid. Figure 10 is an over-all view of the internal pressure test.

During this test, the gages imbedded in the center section of the model (Nos. 13 through 24, Fig. 8) were read to determine how well the measured strains compared to values calculated using conventional formulas for thick-walled pressure vessels. Table 2 shows the comparison between the experimental and calculated tangential strain values.

The gages located behind the lower fillet (Nos. 1 through 12, Fig. 8) were also read during the internal pressure test. It was the authors' original intention to investigate the tangential stress gradient in this area. However, due to a series of broken lead wires only the strain immediately below the fillet was determined. The stress-concentration factor (K_t)† associated with this point, and for this particular manner of loading, was found to be 3.8.

The model was also loaded by applying a static load to the lower flange as shown in Fig. 11. The

† *K_t is defined as the strain measured immediately below the filleted corner, tangent to the fillet radius, at 45 deg to the vertical, divided by the longitudinal strain in the wall at a section removed from the ends.*

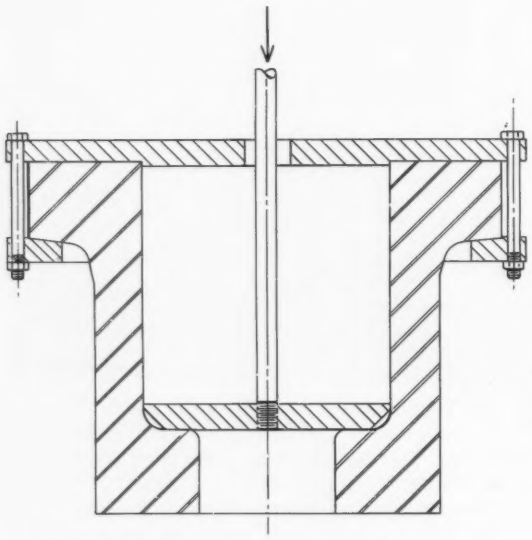


Fig. 11—Static-load test of thick-walled vessel

TABLE 2—TANGENTIAL STRAINS AT MIDSECTION

Gage location	Internal pressure, psi	Measured tangential strain,, μin./in.	Calculated tangential strain, μin./in.
At inside wall	35	170	182
Center of wall	35	134	146
At outside wall	35	115	124
At inside wall	90	495	466
Center of wall	90	372	374
At outside wall	90	280	319

gages behind the upper fillet (Nos. 25 through 40, Fig. 8) were read during this test. The original upper fillet was $^1/_2$ in. The stress concentration factor (K_t) associated with this radius was found to be 4.9. After this value was determined, the vessel was remachined to reduce the radius of the upper fillet to $^1/_4$ in. This new radius is shown by the dotted lines in Fig. 8 and, as shown, by prearrangement a biaxial gage was in position to measure the strain at the base of this new fillet. With a $^1/_4$-in. upper-fillet radius the stress concentration factor was found to be 5.9.

Conclusions and Recommendations

Since the inclusion of the strain elements in the model does not significantly alter the strain pattern within the model, it is thought that this method is well suited to three-dimensional problems. The fact that a single model can be used for a series of tests involving various loading conditions and that the model can be modified between tests appears to be advantageous.

The selection of Castolite for this project was based upon a combination of suitable photoelastic and mechanical properties. However, the use of other materials for further work in this field should be investigated.

Etched foil gages have recently appeared on the market. While these gages may have a slightly greater mass and surface area than the wire gages used in this work, the possibility of their use should be investigated, since this would greatly simplify the task of model making.[5]

During the course of this investigation, several areas in which this experimental method might prove useful have come to the attention of the authors. It will, of course, lend itself to the study of dynamic stress in three-dimensional models.[6] When used in conjunction with temperature-measuring devices, this method might be useful as a tool in thermal-stress studies. Consideration should also be given to the use of the method in conjunction with photoelasticity since the numerical values of strain at interior points may simplify the photoelastic analysis.

Acknowledgments

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California for the Atomic Energy Commission. Appreciation is expressed to Frederick W. DuBois and to Edward J. Sass for their suggestions and assistance in preparing the model.

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Discussion

by S. E. Whitcomb

Brasier and Dove have developed, tested and applied what appears to be an extremely useful technique. The application described by the authors indicates that this technique should be a very useful addition to the methods of studying three-dimensional stress distributions. This technique has the obvious advantages that it is not necessary to destroy the model and that involved calculations are eliminated.

The possible application of this technique to the measurement of stresses under dynamic conditions is of considerable interest. Dove and Williams at the University of New Mexico are currently investigating this possibility. The work reported here shows that the incorporation of the wire strain gage introduces no distortions in the static case. For the study of stress pulses additional requirements must be met and these are now being studied. Dynamic effects may produce distortions in the stress field which do not occur in the static case since, in general, dynamic and static properties may differ.[1] Secondly, the presence of the wire introduces a change in the medium through which stress pulses pass and this may lead to undesirable reflections. It is hoped that the 1 mil wire is sufficiently small that reflection and diffraction effects will be negligible. Thirdly, for stress-pulse studies it is desirable that the length of the gage be small compared to the length of the pulse. The $^1/_4$ x $^3/_8$ x 0.012 in. gage volume described by Brasier and Dove is rather large to study pulses produced by some impacts,[2] and it is hoped that it will be possible to reduce the gage size. If the work which is now under way demonstrates that the imbedded gage introduces no dynamic effects, then this technique should be very useful in the study of the propagation of stress waves.

If its applicability is established, the imbedded-gage technique will be useful in conjunction with the methods of dynamic photoelasticity.[3–7] Imbedded gages would provide a set of calibration points which would greatly simplify the quite laborious calculations which are now required to calculate the stresses as a function of time. By this combination of techniques, it should be possible to obtain valuable information about stress-wave propagation in two dimensions. The imbedded-gage technique when used by itself will yield considerable information concerning wave progagation in three dimensions.

S. E. Whitcomb is associated with Sandia Corp., Albuquerque, N. Mex.

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Authors' Closure

The authors appreciate Dr. Whitcomb's interest. We agree that the work reported in this paper does not demonstrate that the embedded gage can be used to study dynamic strains. As Dr. Whitcomb indicated, we are now working on an extension of this technique to the dynamic case.

Since this paper was submitted, several attempts have been made to imbed commercially available foil gages in Castolite. It appears that if their mass is sufficiently small ($^1/_{16}$-in. gage length or less) they can be used without materially affecting the strain pattern in the model. Larger gages cause proportionally higher residual strains. More recently, strain gages with a thermocouple at the gage point have come to the authors' attention. They, too, have been successfully imbedded in Castolite models. It is hoped that such a strain-gage thermocouple pair can be used in some thermal-stress problems of interest.

Stresses in a Deep Beam with a Central Concentrated Load

The results of a series of photoelastic tests are presented and a comparison between the theoretical and experimental stress distribution is shown on a number of sections

by Shawki Saad and Arnold W. Hendry

ABSTRACT—The stress distribution in deep beams has received the attention of mathematicians and research workers since the early days of Wilson, Stokes and Filon.[1, 2] A general theoretical solution is not possible and particular solutions of the biharmonic equation for the Airy stress function have been worked out to satisfy certain loadings. In the case of a deep beam under a symmetrical pressure on the top edge, solutions by (1) Durant and Garwood,[3] (2) Chow, Conway and Winter[4] have been put forward. Systematic photoelastic experimental work appears to be scarce and the object of this article is to present for comparison with existing theories the results of three photoelastic tests on simply supported deep beams carrying a central point load.

The experimental stresses and those obtained from the solutions referred to above agreed reasonably well. The normal stresses σ_x calculated from the Wilson-Stokes theory were in error for the case under consideration (the theory is not valid for $(H/L) > 0.4$), and the simple theory of bending is adequate for the maximum tensile stress provided the span exceeds 1.5 times the depth.

Results are summarized in figures showing the magnitude and direction of the principal stresses; magnitudes are in a dimensionless form, hence it is possible to adapt them for any beam of similar depth to span ratio under the same type of loading.

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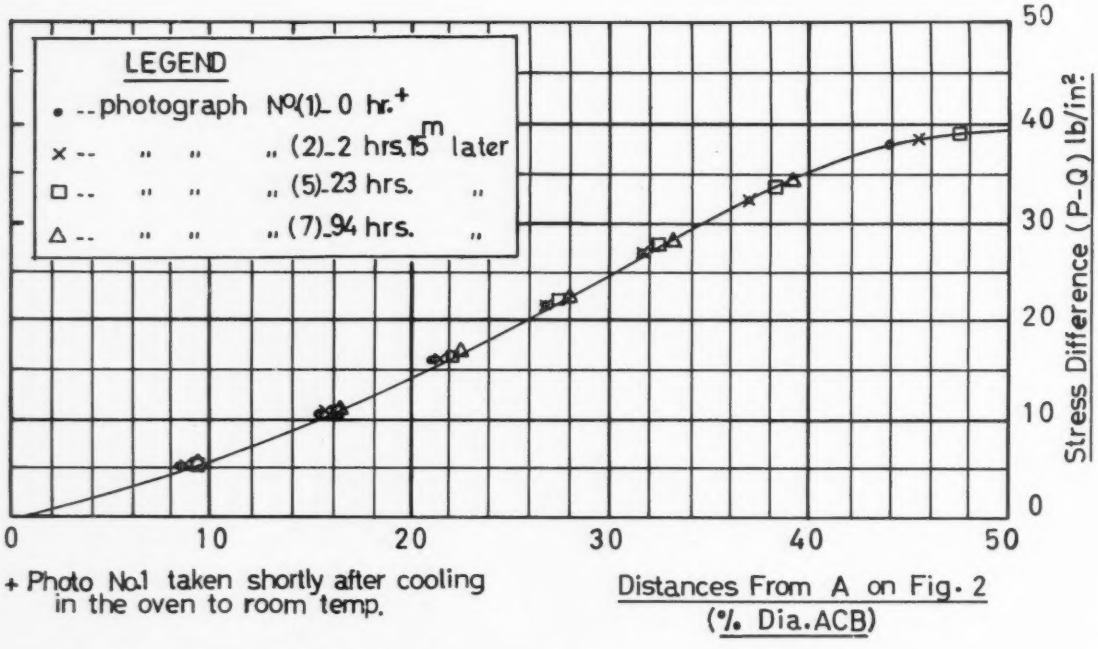
Paper was presented by title at the 1958 SESA Annual Meeting held in Albany on November 12–14.

Introduction

The problem of the stress distribution in deep beams has been investigated theoretically from time to time during the past seventy years and a number of solutions have been produced. Wilson and Stokes,[1] and later Filon,[2] attempted to solve the problem by superimposing a stress field arising from the presence of concentrated loads on the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Stress Difference (P-Q) lb/in²
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Distances From A on Fig. 2
(% Dia. ACB)

Fig. 1—Showing creep effect but no error in (P-Q)

normal bending-stress system in the beam. Later, methods have been based on solution of the biharmonic equation for the Airy stress function; unfortunately, a general solution has not been found and the two methods which have been used to give an approximate solution have been (*a*) by adopting a particular solution to the biharmonic and superimposing a stress system expressed as a polynomial to remove the boundary stresses implicit in the particular solution;[3] (*b*) by numerical solution of the biharmonic using the finite differences method.[4] A certain amount of experimental work has been carried out on the case of a deep beam carrying a central point load but it has been found that some early experiments by Fukuhara, quoted by Coker and Filon[9] and later by Frocht,[10] are inaccurate and that in fact there is a lack of detailed and systematic photoelastic experimental work which would provide a basis for assessing the accuracy of the various theoretical solutions. Unfortunately the photoelastic results by Uhlmann[5] have not been published and those by J. S. Caswell[6] were mainly concerned with the application of the Wilson-Stokes theory and thus were inconclusive. The object of this article is to present the results of a series of photoelastic tests and to show on a number of sections a comparison between the theoretical and experimental stress distribution. It may be added that the accurate calculation of the stresses in a deep beam is rather lengthy and that, by presenting the experimental results in a suitable form, they may be used as a means of assessing the stresses in any beam similar in design.

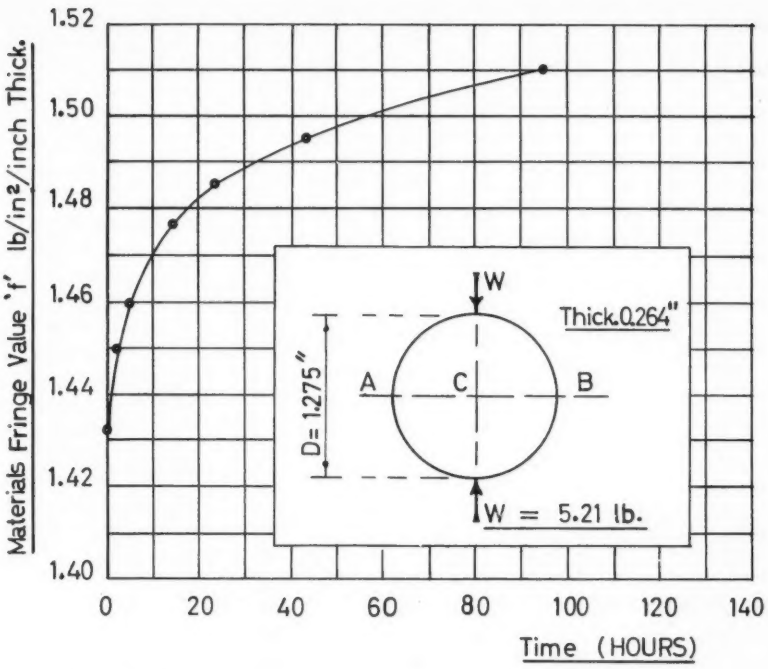

Fig. 2—Change in Catalin fringe value due to creep

Photoelastic Experiments

Normal photoelastic methods were used in the tests and the principal stresses were separated by using the numerical method of solving the Laplace equation for the sum of the principal stresses $(P + Q)$. The grid used for this calculation was arranged so as to avoid the load and reaction points; a few values of $(P + Q)$ in the immediate vicinity of the concentrated loads were, however, determined by calculation using the well-known theory for a line load on a semi-infinite elastic plate.

The material used for the models was a new type of Catalin. This material has to some extent been replaced in photoelasticity by more stable materials, such as Araldite, which show less creep. Catalin, however, has certain advantages in that it possesses a very low fringe value and is very easily machined. In this new form, optical creep, although reduced, is still present; in spite of this, accurate results can be obtained provided that the fringe value is determined at the same length of time after application of the load as in the model under test. This is shown in Fig. 1 in which have been plotted the values of the difference in principal stresses $(P - Q)$ across a diameter perpendicular to the diametrical loading W of the disk in Fig. 2. The stresses were frozen in the usual way and photographs were taken at the time intervals shown on these figures. From similar investigations by Mylonas[7] and Fisher[8] creep in Catalin, which exhibits itself as relaxation in the fringe order, has the same characteristic whether a room-temperature test or the frozen-stress method is adopted. The latter method was preferred because the authors were interested in applying this technique on Catalin models for research on gravitational stresses (paper read at the Fiftieth Anniversary of the Institution of Structural Engineers—London, Oct. 9, 1958). All the experimental points lie on the line calculated by elastic theory, thus confirming that a correct result will be obtained from any of the sets of points, provided that the fringe value is read from the curve shown in Fig. 2.

Experimental Results

Tests were carried out on three simply supported beams in which: (*a*) the ratios of depth H to span L were $^2/_3$:1, 1:1, and 1.59:1; (*b*) the cross section was rectangular; and (*c*) the central point loading and the two reactions were transmitted through steel bars $^3/_{16}$ in. in diameter. Thin rubber pads, nearly 0.04 in. thick, were introduced between the point loading and the top edge of the beams and short aluminum plates, also about 0.04 in. thick were stuck to the models at the supporting points. In the deepest beam, however, no serious effect was noticed when the two supporting bars were bearing directly on the model plate.

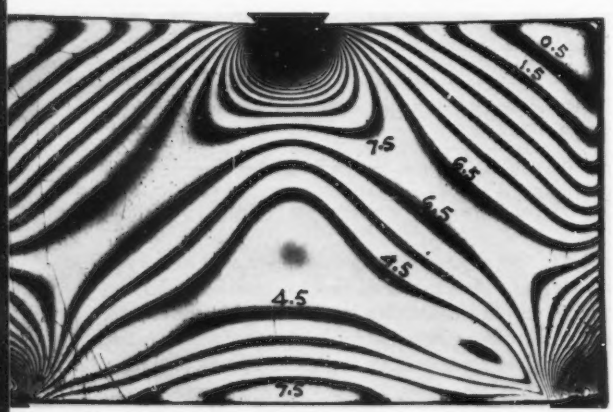

Fig. 3—Stress pattern for a beam of depth (H) = 2/3 span (L)

Figures 3, 4 and 5 show the fringe pattern in each test photographed in mercury-green light with Nicols crossed and quarter-wave plates parallel. Data for the size of the model, the ratio of depth H to span L and the model fringe value at the moment of photographing are indicated in Table 1. The results are summarized in Figs. 6, 7 and 8 which show the magnitude of the principal stresses on a number of sections in each of the beams expressed as a fraction (or multiple) of the maximum stress σ_s in the beam given by the simple theory of bending (i.e., M/Z, where M is the maximum bending moment for a central point load on the span L and Z is the section modulus). The directions of the stresses are also indicated and, from this information, it is possible to calculate the normal, tangential and shear stresses in any section. The dimensionless form of these results facilitates their application to the calculation of the stress in any centrally loaded beam of similar depth-to-span ratio; results for beams of intermediate depth-to-span ratios may be obtained by interpolation. The normal stresses

TABLE 1—DATA FOR TEST MODELS

Fig.	Size of Model ins.	Span (L) ins.	Ht. (H) ins.	Load (P) lb.	Model Fringe Value lb/in.²/ Fringe
3	2.60 × 1.56 × 0.248 thick	2.36	1.56	139.80	95.90
4	2.25 × 1.99 × 0.252 thick	2.04	1.99	124.04	147.10
5	1.73 × 2.36 × 0.252 thick	1.49	2.36	99.59	106.80

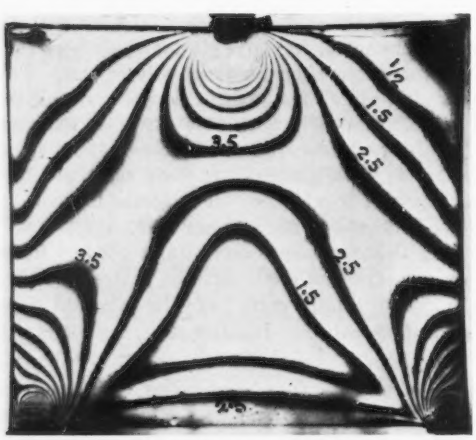

Fig. 4—Stress pattern for a beam of depth (H) = span (L)

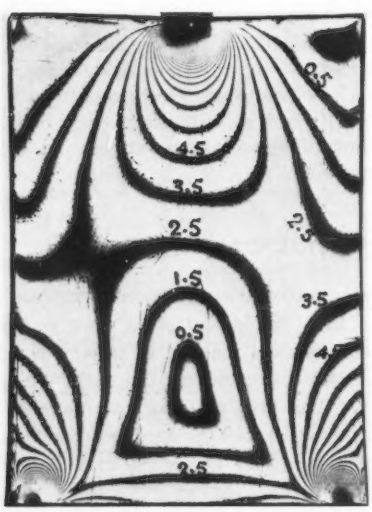

Fig. 5—Stress pattern for a beam of depth (H) =1.59 span (L)

on a section close to the centerline and shearing stresses on three sections are plotted in Figs. 9, 10 and 11; these diagrams give a rapid indication of the extent of the deviation from the simple-beams theory in each case. The photoelastic results were checked statically by calculating from them the bending moments and shearing forces on a number of sections, as indicated, and comparing these with

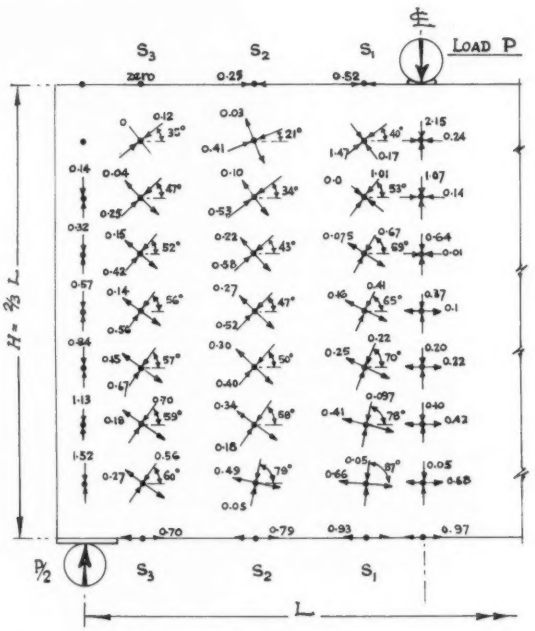

Fig. 6—Principal stresses for beams of depth (H) = 2/3 span (L)

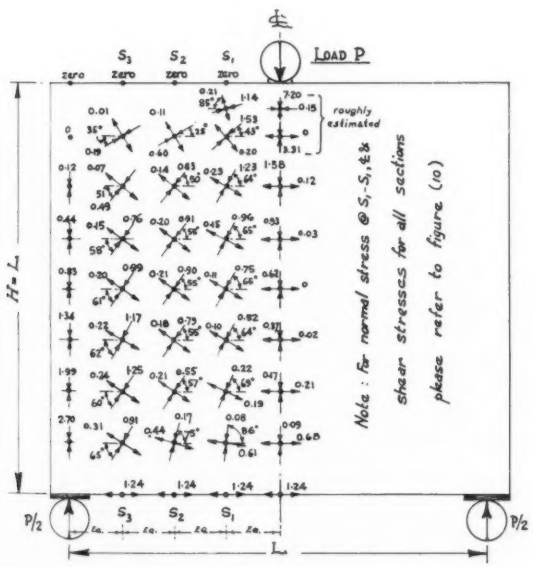

Fig. 7—Principal stresses for beams of depth (H) = span (L)

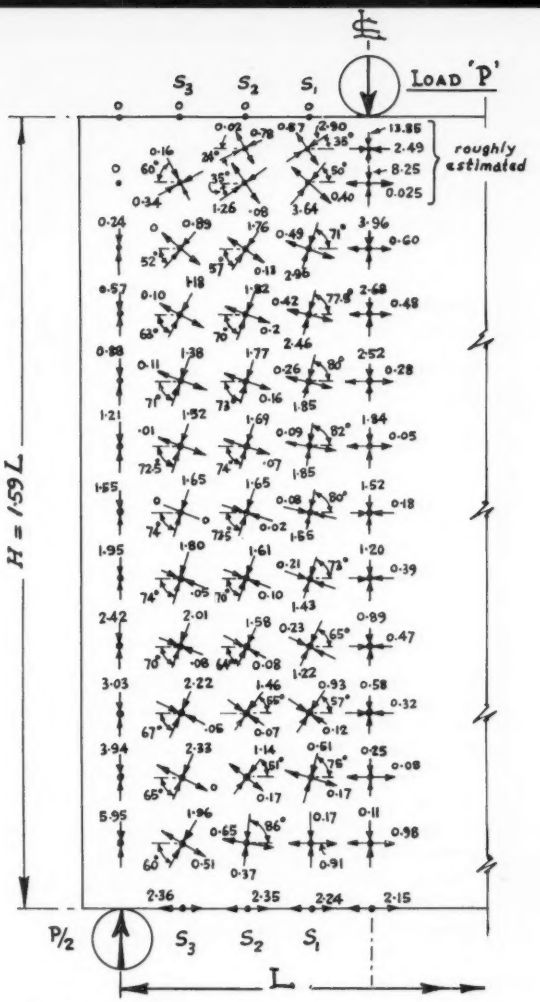

Fig. 8—Principal stresses for beams of depth (H) = 1.59 span (L)

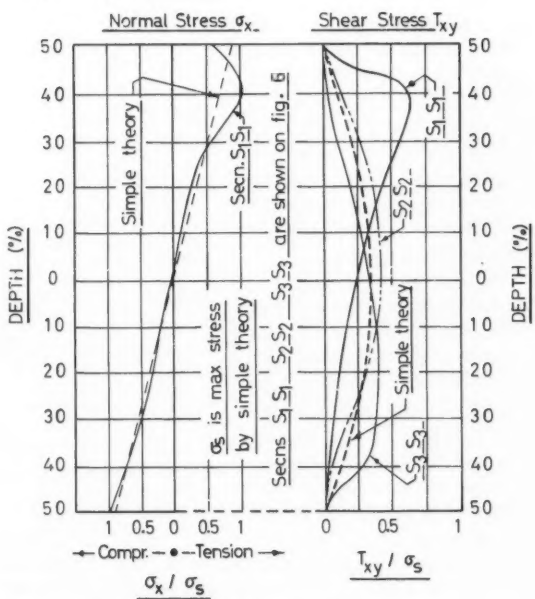

Fig. 9—Normal and shear stresses for beams of depth = 2/3 span

the external bending moments and shear forces. Agreement was found to be correct within a few percent.

Comparison with Theoretical Results

The normal and shearing stresses on certain sections of the model having a depth-to-span ratio of 1:1 have been calculated and are compared with (*a*) the simple theory of bending (M/Z), (*b*) the formal solution of Durant and Garwood[3] (the general equations of stresses are given in the Appendix) and (*c*) the numerical solution by finite differences worked out by Chow, Conway and Winter.[4] These comparisons are shown in Fig. 10. Two curves of the experimental normal stresses σ_x are given, of which the complete curve along section S_1S_1 is plotted mainly for the statical check on bending. The difference in the experimental stresses σ_x between the centerline and section S_1S_1 is small along the lower half of the depth H, but near the top edge the compression on the latter falls rapidly to zero at the boundary. Photographs in Figs. 3, 4 and 5 show that the compression along the top edge and away from the point loading is zero except for the case in which H/L is two-thirds.

Similar agreement between the experimental and the theoretical results was obtained for beams having greater and smaller ratios of H/L and when this ratio is less than one-half the experimental tensile stresses σ_x are closely represented by the simple bending theory.

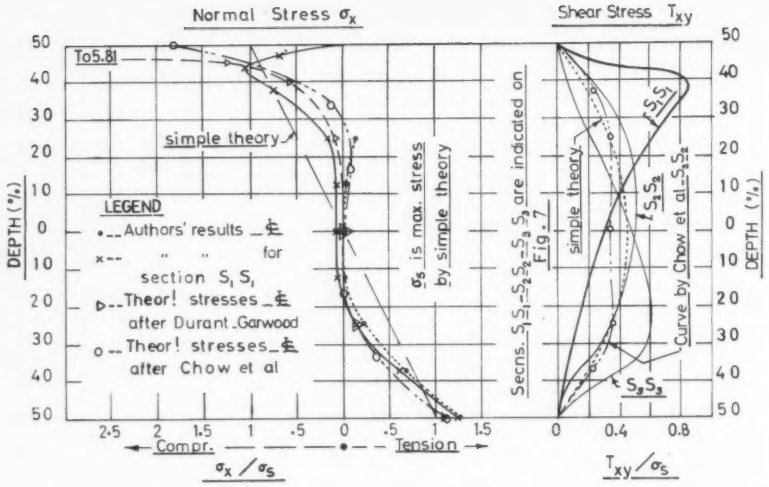

Fig. 10—Normal and shear stresses for beams of depth = span

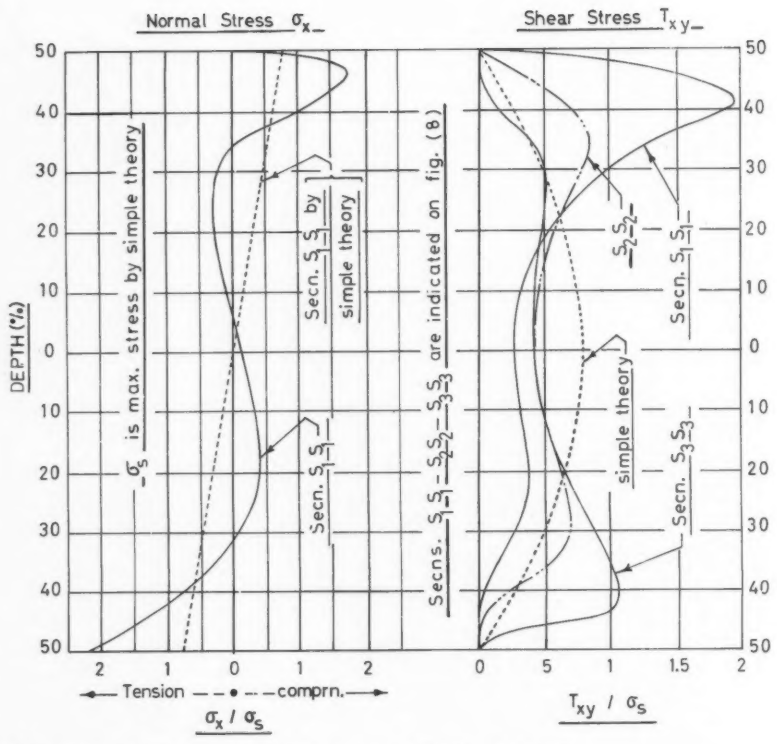

Fig. 11—Normal and shear stresses for beams of depth = 1.59 span

A further comparison is given in Fig. 12 which shows the ratio of the maximum tensile stress on the lower edge of the beam to the corresponding stress given by the simple theory of bending, plotted against the ratio H/L. Also shown is the curve given by the Wilson-Stokes theory which has been quoted by Coker and Filon[9] and by Frocht,[10] along with some early results given by Fukuhara; the

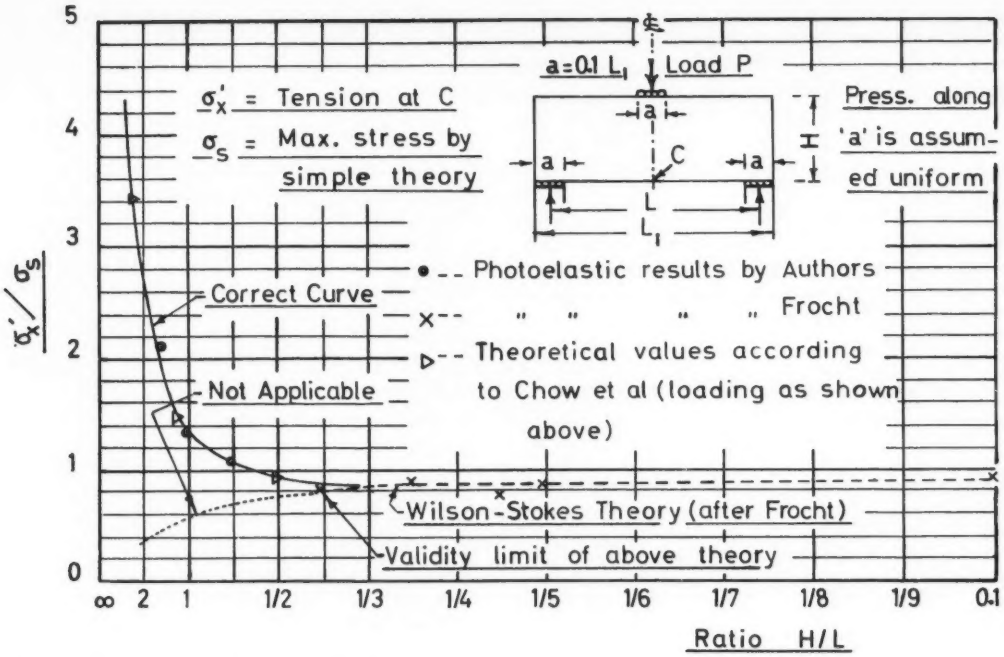
a=0.1 L1
Load P
σx' = Tension at C
σs = Max. stress by simple theory
Press. along 'a' is assumed uniform
Photoelastic results by Authors
" " " " Frocht
Theoretical values according to Chow et al (loading as shown above)
Correct Curve
Not Applicable
Wilson-Stokes Theory (after Frocht)
Validity limit of above theory
σx'/σs
Ratio H/L

Fig. 12—Validity limit of Wilson—Stokes theory

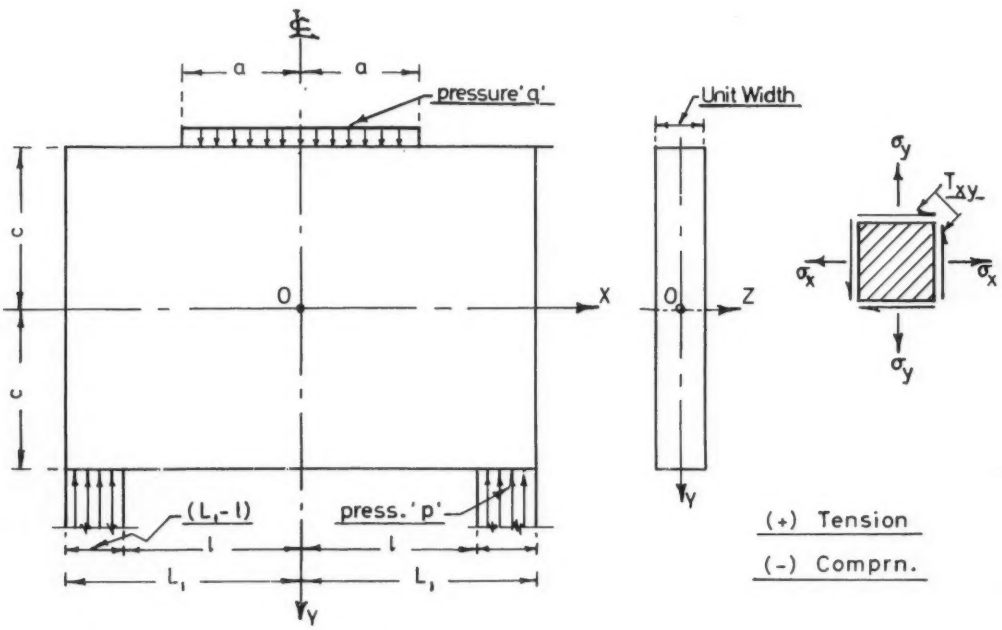

Fig. 13—Notations adopted and used by Durant—Garwood

results obtained by Fukuhara came from photoelastic tests on celluloid beams and are evidently seriously in error.

Conclusions

The results presented in this article provide substantial confirmation for the solutions for a centrally loaded deep beam put forward by (1) Durant and Garwood, (2) Chow, Conway and Winter using the finite differences method. They may also be employed directly for the estimation of stresses in deep beams. The following observations may be

made on the basis of these results:

(*a*) The normal stresses given by the theories referred to above are reasonably accurate but high stresses obtained in the immediate vicinity of the load point may be neglected; the stress along the top edge of a beam of depth to span ratio of 1:1 and over is zero except under the load where it is indeterminate.

(*b*) The Durant-Garwood solution is probably the most rapid method if stresses are required only on one or two sections. The finite differences method may be better if the stresses over the whole beam are required but the arithmetical work is very lengthy.

(*c*) The theoretical solutions will become very difficult or even impracticable, if the loading is unsymmetrical or if there are holes in the beam. In this case the easiest solution is by photoelastic analysis.

(*d*) The Wilson-Stokes theory is misleading and should not be used for the calculation of normal stresses where the span is less than 2.5 times depth.

(*e*) The simple-bending theory is adequate in the case under investigation for beams in which the span exceeds one-and-a-half times the depth.

Appendix

N.B. : q & p were assumed = pt. load & reaction/0.1L_1 resp.

Referring to fig. 13 and denoting

$$\alpha = m\pi/L_1 \quad , \quad \beta = \alpha c \quad , \quad \alpha x = \xi \quad , \quad \alpha y = \eta \quad ,$$

$$\beta - \eta = u \quad , \quad \beta + \eta = v \quad \text{and}$$

$$K = \frac{L_1^2}{4C^3}\left\{p\left[\frac{l}{L_1} - \left(\frac{l}{L_1}\right)^3\right] + q\left[\frac{a}{L_1} - \left(\frac{a}{L_1}\right)^3\right]\right\} \quad , \quad \text{then}$$

$$\sigma_x = \left\{ Ky + \left[\sum_{m=1}^{\infty} h_m \cos\xi \Big/ (\sinh^2 2\beta - 4\beta^2)\right] \times \right.$$
$$\left[\sinh 2\beta (u\cosh v - \sinh v) - 2\beta(v\cosh u - \sinh u)\right]$$
$$+\left[\sum_{m=1}^{\infty} a_m \cos\xi \Big/ (\sinh^2 2\beta - 4\beta^2)\right] \times$$
$$\left.\left[\sinh 2\beta(v\cosh u - \sinh u) - 2\beta(u\cosh v - \sinh v)\right]\right\}$$

$$\sigma_y = \frac{1}{2L_1}\left[qa + p(L_1 - l)\right] - \rho g x y - \left[\sum_{m=1}^{\infty} h_m \cos\xi \Big/ \right.$$
$$(\sinh^2 2\beta - 4\beta^2)\Big]\Big[u\sinh 2\beta\cosh v - 2\beta v\cosh u +$$
$$\sinh 2\beta\sinh v - 2\beta\sinh u\Big] - \left[\sum_{m=1}^{\infty} a_m \cos\xi \Big/ \right.$$
$$(\sinh^2 2\beta - 4\beta^2)\Big]\Big[v\sinh 2\beta\cosh u - 2\beta u\cosh v +$$
$$\sinh 2\beta\sinh u - 2\beta\sinh v\Big]$$

$$\tau_{xy} = \sum_{m=1}^{\infty}\left[h_m \sin\xi \Big/ (\sinh^2 2\beta - 4\beta^2)\right] \times (u\sinh 2\beta \times$$
$$\sinh v + 2\beta v\sinh u) - \left[\sum_{m=1}^{\infty} a_m \sin\xi \Big/ \right.$$
$$\left.(\sinh^2 2\beta - 4\beta^2)\right] \times (v\sinh 2\beta\sinh u + 2\beta u\sinh v)$$

wherein :

$$a_m = 2q\sin\alpha a/m\pi \quad , \quad h_m = -2p\sin\alpha l/(m\pi)$$

& ρg = Density of material

Addendum

The article by P. H. Kaar[11] was brought to the knowledge of the authors just after this paper was prepared for publication. The experimental work by Kaar was carried out on limited sizes of steel and aluminum plates and electric-resistance strain gages were used. These gages need space, and measurements had to be confined to the centerline of the beams; the advantage of photoelastic analysis in such cases is apparent. Agreement between dimensionless results of Kaar's experiments and those by the authors is reasonable within the permissible experimental error, and so are his conclusions, except for the last statement regarding beams with $H/L > 2$, which were not included in the authors' tests.

Acknowledgment

The experimental work referred to in this article was carried out in the Faculty of Engineering, University of Khartoum; thanks are due to the University authorities for the facilities provided.

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Strain-gage Measurements in Regions of High Stress Gradient

Technical note presents some of the data collected during measurement of maximum strain in regions of rapidly varying strain

by C. S. Ades and L. H. N. Lee

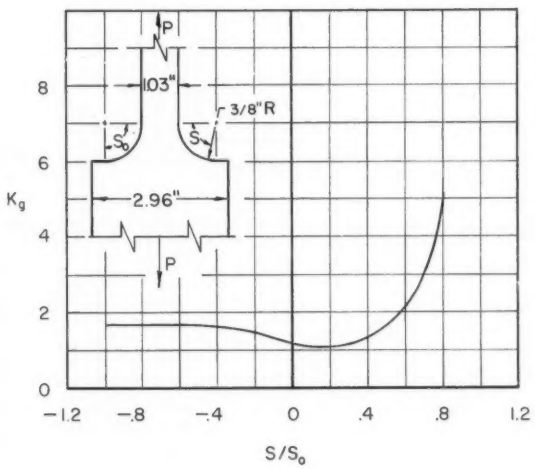

Fig. 1—Error in maximum strain measurement due to gage location. Gage length = 1/8 in. K_g = max. strain/measured strain

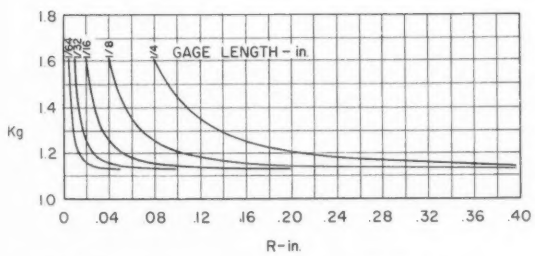

Fig. 2—Error in maximum strain measurement due to gage length—mid-point of gage at point of tangency near point of maximum stress

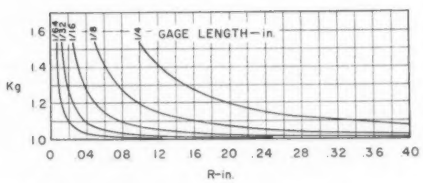

Fig. 3—Error in maximum strain measurement due to gage length—mid-point of gage at point of maximum stress

In attempting to measure the maximum strain with a resistance-type gage in a region where the strain is varying rapidly, there are two problems which face the experimental analyst: first, where must the gage be located; and, secondly, what length gage should be employed? The writers were once asked to analyze some data obtained by the use of strain gages in regions of high stress gradient, and arrived at some answers with the help of photoelasticity. Presented herein are some of the data derived as a result of this investigation.

Figure 1 illustrates the deviation of the gage reading from the maximum strain for a strain gage with its centerline located at various distances from the point of tangency of a filleted bar. K_g equals the maximum strain divided by the gage reading. Although the data was computed for a 1/8-in. gage and a 3/8-in. radius, it is presented in nondimensional form. It should be noticed that the maximum does not occur at the point of tangency, but occurs at the radius inclined 11 deg with the radius through the point of tangency. This agrees with information presented by Heywood.[1] Three-dimensional tests,[2] however, indicate that the point of maximum strain is not fixed, but can occur at various points on the 11 deg arc of the fillet adjacent to the point of tangency.

Figure 2 illustrates the error in the determination of the maximum strain if one uses gages of various lengths in fillets of various radii. Here the centerline of the gage coincides with the point of tangency. Figure 3 gives similar information except that here the centerline of the gage coincides with the point of maximum strain. This graph illustrates the strain sensing errors due to length alone. It also shows that these errors are not excessive—that is, they are not greater than normal instrumentation errors—if the gage length is less than one-third the length of the radius. While the data presented here were all derived from the same filleted configuration, the authors have worked out similar data for widely varying filleted configurations, and the results obtained have exhibited the same general characteristics.

From the above, it is seen that considerable error can occur in using a single gage to determine a maximum strain. It is thus necessary to seek an alternate approach. Photoelasticity would be the logical

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answer, but the equipment, loading systems, models and ample time are not always available. If a test involving the prototype structure and correctly simulated loading conditions can be performed, one approach would consist of taking strain readings from several gages. A method of determining the maximum strain based upon multiple gage readings is presented in the following paragraphs.

The strain distribution along a line on the surface of a deformed body may be described by a mathematical function such as $\epsilon(x)$. If the function possesses derivatives of all orders at a point $x = x_0$, then it may be expanded into a Taylor's series about the point x_0. Consider that interval $x_1 < x < x_2$ and that $x_1 < x_0 < x_2$. If the interval is sufficiently small, the strain distribution may be approximated by a line designated by the first two terms of the series, which is obviously only good for this particular interval. When the interval is large, more terms of the series would be needed to provide a description of the actual distribution with suitable accuracy. If the location of maximum strain is known to be within a certain interval, and that interval is sufficiently small, the strain distribution may be approximated by the first three terms of the series, which can be written in the following form:

$$\epsilon(x) = a + bx + cx^2 \qquad (1)$$

It is evident that only one maximum strain can be within this interval. Let us consider that three gages of equal length may be placed in the interval in which eq. 1 is valid as shown in Fig. 4. The gages may or may not be overlapped. Now if ϵ_1, ϵ_2 and ϵ_3 are the three gage readings, then

$$\epsilon_1 = \frac{1}{L}\int_0^L \epsilon(x)dx = a + \beta_1 b + \gamma_1 c$$

$$\epsilon_2 = \frac{1}{L}\int_{\delta_2}^{L+\delta_2} \epsilon(x)dx = a + \beta_2 b + \gamma_2 c$$

$$\epsilon_3 = \frac{1}{L}\int_{\delta_3}^{L+\delta_3} \epsilon(x)dx = a + \beta_3 b + \gamma_3 c \qquad (2)$$

where

$$\beta_1 = L/2, \quad \beta_2 = (L + 2\delta_2)/2, \quad \beta_3 = (L + 2\delta_3)/2$$
$$\gamma_1 = L^2/3, \quad \gamma_2 = (L^2 + 3L\delta_2 + 3\delta_2^2)/3, \quad \gamma_3 = (L^2 + 3L\delta_3 + 3\delta_3^2)/3 \qquad (3)$$

Solving the three equations simultaneously yields the following:

$$a = [(\beta_2\gamma_3 - \beta_3\gamma_2)\,\epsilon_1 + (\beta_3\gamma_1 - \beta_1\gamma_3)\,\epsilon_2 + (\beta_1\gamma_2 - \beta_2\gamma_1)\,\epsilon_3]/\Delta \qquad (4)$$

$$b = [(\gamma_2 - \gamma_3)\,\epsilon_1 + (\gamma_3 - \gamma_1)\,\epsilon_2 + (\gamma_1 - \gamma_2)\,\epsilon_3]/\Delta \qquad (5)$$

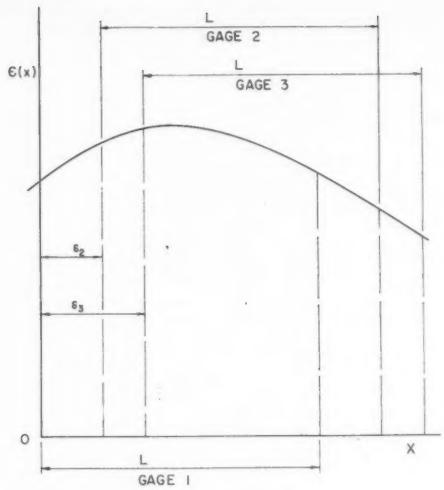

Fig. 4—Strain distribution and assumed gage locations with multiple gages

$$c = [(\beta_3 - \beta_2)\,\epsilon_1 + (\beta_1 - \beta_3)\,\epsilon_2 + (\beta_2 - \beta_1)\,\epsilon_3]/\Delta \qquad (6)$$

$$\Delta = (\beta_2\gamma_3 - \beta_3\gamma_2) + (\beta_3\gamma_1 - \beta_1\gamma_3) + (\beta_1\gamma_2 - \beta_2\gamma_1) \qquad (7)$$

With a, b and c known, there is available a close approximation of the strain distribution covered by the gages. The maximum or minimum point can be found by equating the first derivative to zero:

$$\frac{d\epsilon(x)}{dx} = b + 2cx = 0 \quad \text{or} \quad x = -b/2c \qquad (8)$$

By taking the second derivative, it can also be readily shown that the solution is a maximum if c is negative. Substituting eq 8 into eq 1, it is found that the maximum strain

$$\epsilon_{max} = a - b^2/4c \qquad (9)$$

If more accuracy is needed, four gages can be used—the equations are derived in a similar manner. For a strain gage of sufficiently small length, L, however, the second order approximation as given should provide a result with an error less than the probable experimental errors in most cases.

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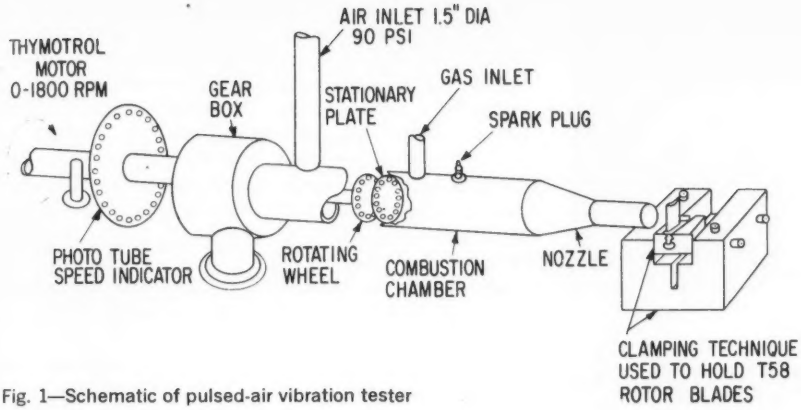

Fig. 1—Schematic of pulsed-air vibration tester

Pulsed-air Vibration Technique for Testing High-performance Turbomachinery Blading

Paper discusses the design and operation of a pulsed-air vibration exciter which has proved to be an effective laboratory tool for blade-vibration studies

by J. C. Truman, J. R. Martin and R. V. Klint

ABSTRACT—A pulsed-air vibration exciter for testing turbomachinery blading is described. This exciter has been used successfully in performing blade-vibration studies at frequencies up to 15,000 cps. Higher mode fatigue failures, both flexural and torsional, have been produced. Although capable of vibrating large blades, the exciter has been particularly useful in investigations of small blades having inherently high natural frequencies. The tester is capable of high-temperature operation.

Operating experience using the pulsed-air exciter is presented, and examples of its application to engine-design problems are given.

The Problem

The aircraft gas turbine, early in its history, proved itself to be an effective blade-fatiguing device. Accordingly, it soon became apparent to designers that vibration and fatigue considerations must play an important role in blade-design criteria. It was also recognized that improved laboratory techniques for blade vibration and fatigue testing were required.

As jet-engine design became more refined, the trend progressed toward lighter, more highly stressed blading, and blade-vibration problems became more critical and complex. Analytical techniques and vibration analyses became recognized as fundamental to good blade design.

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Analysis, alone, however, is not always enough:

1. Assumptions which are necessary for mathematical simplification may not be completely realistic, and a more exact analysis is often too time consuming and costly.
2. Strain-gage response at a given blade location may not be readily predictable, making testing necessary for optimum gage location, and calibration desirable for increased stress-monitoring accuracy.[1]
3. Blade quality, materials and manufacturing techniques may be readily evaluated by testing, using blade frequencies and fatigue characteristics as indices.
4. Testing further provides a valuable feedback to correlate with design predictions, and establishes a basis for design changes, as well as a means of evaluating such changes.

Specifically, blade vibration testing can be categorized as follows:

1. Determination of natural frequencies.*
2. Determination of spanwise gage response for optimizing strain-gage location for various critical modes.
3. Calibration of strain gages at specific locations on the blade for various modes in preparation for engine monitoring.
4. Fatigue testing to determine blade strength levels.

* *It is, of course, recognized that laboratory results obtained in the absence of a centrifugal field must be corrected to represent engine operating conditions.*

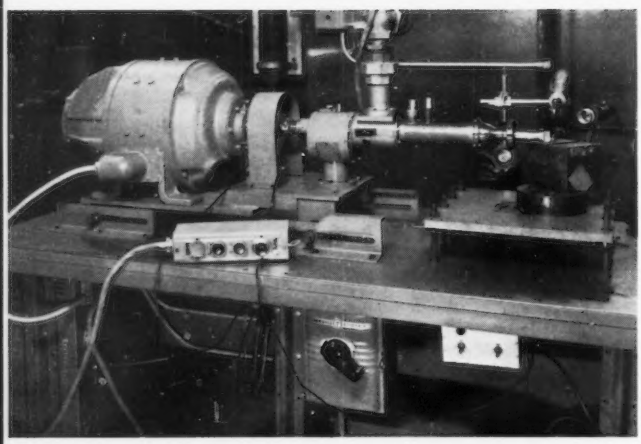

Fig. 2—Photograph of pulsed-air vibration tester

This paper discusses the design and operation of a pulsed-air vibration exciter which has proved to be an effective laboratory tool for blade-vibration studies. The exciter is a siren-type device, comprised basically of a rotating perforated disk which "chops" an air jet directed against the blade. It is capable of vibration testing at high temperature, and can provide high vibratory energy at frequencies up to 15,000 cycles per second.

Vibration Excitation Methods

Various methods of exciting turbomachinery blade vibrations are available to the engineer. These have merit and limitations depending on the application and information desired. Some of these methods are as follows:

1. Electromagnetic excitation. The blade is vibrated by means of a variable-frequency electromagnet, often with a d-c bias to improve the magnetic circuit. This method is particularly useful for obtaining resonant frequencies and mode shapes. However, it is usually difficult to get sufficient vibratory energy, particularly in higher modes with small blades, to produce fatigue failure. Further, this technique is obviously limited to magnetic materials. To overcome this limitation, magnetic masses may be mounted on the blade but these plainly alter the mode shapes, stress distributions and resonant frequencies. The electromagnetic technique, although it produces some electrical heating of the blade, cannot readily be applied to investigating blades at high temperature. For small blades at high frequencies, stray magnetic flux can make strain-gage measurements difficult.

2. Mechanical flexing. The blade may be flexed by a mechanical means, such as a universal fatigue testing machine, or a vibration shake table. While the forces may be large enough to produce fatigue failures and the blades may be tested at high temperature, the vibratory frequencies are usually well below the first flexural frequencies of the blades. Due to the method of application of the force, the stress distribution is usually altered from that to be expected in the engine. In addition, the low excitation frequency results in a long test.

3. Mechanical excitation. The blade is driven by a variable-frequency force usually applied to the blade mount. This force may come from an electromagnetic vibration shake table, or a piezoelectric crystal such as barium titanate, or a magnetostrictive metal bar. The blade may be heated by an oven, if desired, and resonant frequencies and mode shapes may be studied. However, at higher frequencies, excitation forces are usually too small to enable fatigue stresses to be obtained.

4. Jet-nozzle excitation. A high-velocity air jet is trained on the blade end. This ordinarily only excites the first flexural mode of the blade; other modes are difficult or impossible to excite. The vibration may be rather unstable.

5. Tuned air chambers. In conjunction with an air jet impinging on the blade, tunable air chambers may be introduced into the air circuit adjacent to the blade. For best results, the open end of the duct should be shaped to fit the blade contour. While it is possible to excite higher modes of vibration of the blade, the dominance of the first flexural mode as the air chambers are tuned limits this method largely to this mode. Since the blade is an integral part of the aerodynamic circuit, the spacing of the blade and jet nozzle is critical and limits the maximum tip deflection obtainable.

These techniques are generally known and used in laboratory vibration studies. References 2, 3 and 4 present some of these methods in greater detail. An additional technique employing an interrupted air jet has been used by Kemp, Morgan and Manson[5] in determining strain distributions of hollow turbine blades in various modes.

The Pulsed-air Vibrator

It can be seen that several criteria must be met by the technique devised to produce fatigue in turbomachinery blades.

1. Excitation frequency should be variable up to about 15,000 cps.

2. The excitation should be adequate to produce fatigue stresses at resonance in blades over the whole frequency range.

3. The source of excitation should be sufficiently independent of the blade response to permit any mode of vibration within the frequency range to be excited.

4. The excitation frequency, once set, should remain fixed permitting continued running at the particular resonant frequency chosen.

5. There should be provision for high-temperature testing.

6. There should be no restrictions on the material of the blades (e.g., magnetic).

7. The technique must accommodate a wide range of blade size and shape.

8. There should be no mechanical attachment to the airfoil, either in the form of push-rods or magnetic masses.

The pulsed-air technique described here was devised to meet these criteria.[3] As can be seen in Figs. 1 and 2, air is passed through a slotted rotating wheel which is close to a stationary plate having identical openings. This alternate opening and closing of the duct produces pulsations in the air. The frequency of pulsations is proportional to the speed of rotation and number of slots. The pulsating air stream travels down a duct and impinges on the blade mounted at the end of the duct. If high-temperature testing of the blade is desired, combustible gas is added and burned in the duct. The rotating wheel is driven by a variable speed d-c motor through a speed-increasing gearbox.

The rotating wheel is somewhat over two inches in diameter and $^1/_8$ in. thick at the air passage. Twenty slots about $^1/_8$ by $^1/_4$ in. are equally spaced near the rim of the wheel. Since the stationary plate is identical to this wheel, a wheel speed of 45,000 rpm is required to obtain the pulse frequency of 15,000 cps. A 3-hp General Electric Thy-mo-trol variable-speed d-c motor and controller, together with a gearbox supplying the required speed change, is used to drive the wheel. This permits the frequency to be changed continuously from 750 to 15,000 cps using a 90–1800 rpm motor and a 25:1 gearbox. For lower frequency studies, the gearbox can be changed to 5:1 ratio and a 10-slot wheel-and-plate combination used; the frequency range is then 75 to 1500 cps. Excitation frequency is obtained by interrupting a phototube light circuit with a separate slotted wheel mounted on the motor shaft.

Early experience indicated that a specially made "shiftable" gearbox with 25:1 and 5:1 ratios allowed too much backlash, causing fluctuations in the generated frequency. By eliminating the ability to "shift gears," backlash was reduced sufficiently to allow steady vibrating stresses to be obtained. Separate gearboxes of the required ratios were therefore used. However, by changing the wheel-and-plate combination (relatively easily done) the frequency range with the 25:1 gearbox was then 375 to 15,000 cps, which satisfied needs for most of the tests conducted. A flywheel mounted on the output shaft of the gearbox also reduced frequency fluctuations.

The vibration amplitude of the blade is controlled by a valve on the air supply. It was found best to set a small amount of air flow through the "chopper" sufficient to allow strain-gage (or telescope tip-amplitude) readings to be noted as the frequency varied. Once the resonant frequency is obtained and set by the Thy-mo-trol control, the air flow is increased until the desired stress level in the blade is obtained. A slight adjustment of the speed control can be made at this time to insure that the excitation frequency is precisely at the blade resonant frequency, thus obtaining the maximum vibration amplitude.

The Thy-mo-trol is sufficiently stable so that the blade remains in resonance with little or no adjusting of the speed control. Actually, the resonant frequency of the blade may change by a few cycles per second during the first few minutes of running, as the dovetail seats; this requires more adjustment of the speed control than is needed to correct for any "drift" of the Thy-mo-trol.

The intensity of the pulsations produced is a function of the spacing between the rotating wheel and the stationary plate. It is desired to minimize the spacing while insuring that the two do not rub. A clearance of one to five mils has been found adequate to be sure that the surfaces do not rub while still producing pulsations of adequate intensity. These pulsations travel essentially at sonic velocity down the duct to the blade while the air mass moves at a considerably slower speed, depending upon the amount of air flow. It is the pulsations that produce the blade vibration, and not the air flow. Hence, for a given clearance between the wheel and plate, increasing the air flow increases the pressure drop across the wheel and plate, and increases the amplitude of the pulsations.

Measurements of the aerodynamic excitations at the Research Laboratory have indicated that pressure pulsations of about 0.7 psi (peak-to-peak) can readily be obtained with this apparatus. The wave form of the pressure pulsation is triangular, becoming sawtooth as distance from the source is increased. This is shown by photographs in Reference 7. The actual motion of the blade at resonance is essentially sinusoidal; Fig. 3 is an oscilloscope trace of a typical strain-gage signal.

It was also found at the Research Laboratory that gaseous fuel could be added and burned after the pulsations had been produced. This eliminated the very difficult problem of "chopping" preheated air when high-temperature operation was desired. Hydrogen was found to be quite satisfactory as fuel because of its very wide flammability limits (permitting operation from 300° F to over 2000° F) and because of its innocuous combustion products. In operation, the fuel was injected a few inches downstream from the stationary plate. A simple, open-ended tube injecting fuel in the center of the duct initially showed the feasibility of this form of air heating, but flow instabilities caused a "noisy" combustion and hence fluctuations in the amplitude of the pulsations. This was overcome by using an air-foil-shaped tube down the center of the duct, with numerous escape holes. Another satisfactory solution was to supply the hydrogen to the outside edge of the duct by means of an annular manifold. In all cases, a spark plug about an inch downstream from the fuel injector initiated combustion. The spark plug was not needed once combustion was started. By using a duct about 12 in. or more in length, air was well mixed after burning. Attenuation of the pressure pulsations due to the duct was not notice-

able. This was shown by using a series of shorter ducts; the vibration of a given blade with fixed air-flow and wheel-plate clearance was found to be about the same.

Operating Experience

The pulsed-air vibration tester has been used extensively in blading design and evaluation programs on three specific engine projects of General Electric's Small Aircraft Engine Department: the T58, the J85 and the T64. Each of these engines has blades ranging from approximately one half to four inches in length. Significant testing has also been performed on larger blading, particularly in higher mode investigations for which available electromagnetic equipment was not adequate. The largest blade actually tested to date was approximately ten inches long, with a chord of around three inches.

A design problem in which the pulsed-air tester proved extremely valuable occurred in development testing the T58 engine.[8] After some twenty-five hundred hours of engine testing were accumulated in the development program, several fatigue failures occurred in the dovetails of the 8-, 9- and 10-stage rotor blades of two engine assemblies. From analysis of the stress and vibration characteristics of the blades and dovetails, vibration in the first torsional mode was suspected. Since the engines on which these failures had occurred had not been stress-instrumented, it was not possible to confirm the analysis directly from engine-test data. However, these failures were duplicated in the laboratory, providing data on which appropriate design changes could be based.

Blades were subjected to vibration in the first torsional mode using the pulsed-air tester, and dovetail failures closely resembling those experienced on engien test were achieved. This similarity can be seen in Fig. 4, which shows photographs of an engine failure (bottom) and a failure induced in the laboratory (top). The dovetails were then redesigned to provide increased torsional-fatigue strength, and evaluation of the redesigned dovetails was performed on the pulsed-air tester. The effectiveness of the redesign, as indicated by the favorable results of these fatigue tests, has since been confirmed by more than eighteen thousand hours of failure-free engine operation.

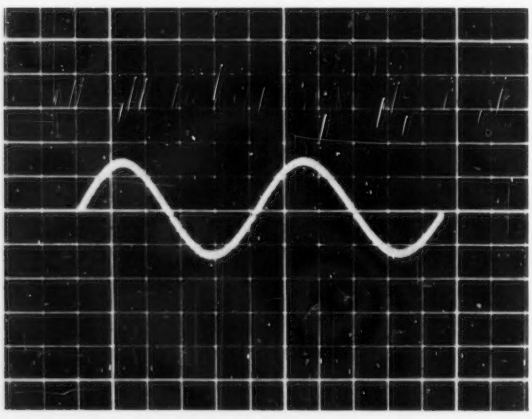

Fig. 3—Oscilloscope trace of strain-gage signal from blade vibrating in pulsed-air tester

The pulsed-air tester has been particularly useful in fatigue testing small blades in higher modes. Figure 5 shows four examples. The first blade (from left to right) was fatigued in the first torsional mode at 9500 cps. The other three blades were fatigued in the second flexural mode at 5000, 1900 and 4540 cps, respectively.

Operating Techniques

Like many pieces of laboratory equipment, the pulsed-air vibration tester has a "personality" with which the experimenter must become familiar if he is to achieve its full potential. It has been found necessary to adjust and monitor carefully such factors as air flow, relative location (axially, chordwise and spanwise) of the nozzle with respect to the blade, and angle of attack of the blade to the air jet. Rotor speed must be controlled carefully, particularly when the blade has a sharp response peak. Nozzle size and shape must be experimented with for some blades to achieve best results. As in all bench vibration testing at resonance, careful attention must be given to minimizing damping in the holding fixture. In several cases, difficulties in fatiguing blades, especially in higher modes, have been eliminated by tighter clamping or improved fixtures.

It is, of course, necessary to provide adequate protection from the high noise levels which are generated, both for operating personnel and for personnel working in adjacent areas. The latter has been accomplished by locating the tester in a remote area, or in an acoustically insulated engine test cell. Operating personnel are protected by ear plugs and acoustic helmets. Recently a sound-absorbing box has been designed and built to mount directly on the base of the machine, covering the blade being tested. The air is exhausted through a series of baffles. Windows in the box are provided for observation of the vibrating blade, and for optical measurement of blade amplitude.

A characteristic of the pulsed-air vibration tester which should be recognized is the relatively large steady-state component of aerodynamic loading to which the blade may be subjected. Thus, the blade is vibrated not about its normal equilibrium position, but about a deflected position, and measured endurance limits are actually those under combined steady and vibratory loads. To detect this, the tip of the blade is marked to reflect a light source, and the bright spot is zeroed on the symmetrical scale of the monitoring telescope. If the amplitude as observed through the telescope is not symmetrical

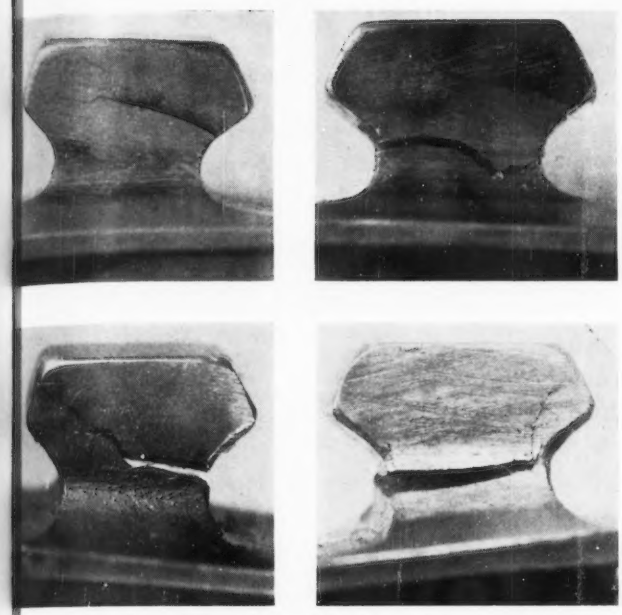

Fig. 4—Bottom: right and left faces of dovetail which failed during engine test. Top: right and left faces of similar dovetail fatigued in torsional mode by pulsed-air tester; frequency 8060 cps

about the zero position, a significant steady load is present. Repositioning the blade with respect to the nozzle will usually reduce the steady load to a negligible value, so that the vibratory strength alone is determined. In any event, steady-load stresses can be calculated readily, and the resultant-stress picture is obtained by superposition of alternating and steady stresses on a modified Goodman diagram.

Caution must be exercised by the operator in controlling the air flow against the blade. This is particularly true for small blades; at their characteristically high frequencies, normal "runout" occurs quickly, and the operator can afford little time for initial adjustments. (For example, at 8000 cps, 10^7 cycles take only about 21 minutes.)

Acknowledgments

Several individuals of the General Electric Co. have contributed to the design of the pulsed-air vibration tester, and to the development of experimental techniques which have made it an effective laboratory tool. The authors acknowledge the contributions of S. E. Gamarekian, R. E. Tuft, R. S. Owens, and O. Brusdal of the Research Laboratory; C. Crocker Jr., L. J. Joyce, G. D. Kouloheras, C. E. Peaslee and C. M. Smith of the Small Aircraft Engine Department, Flight Propulsion Division.

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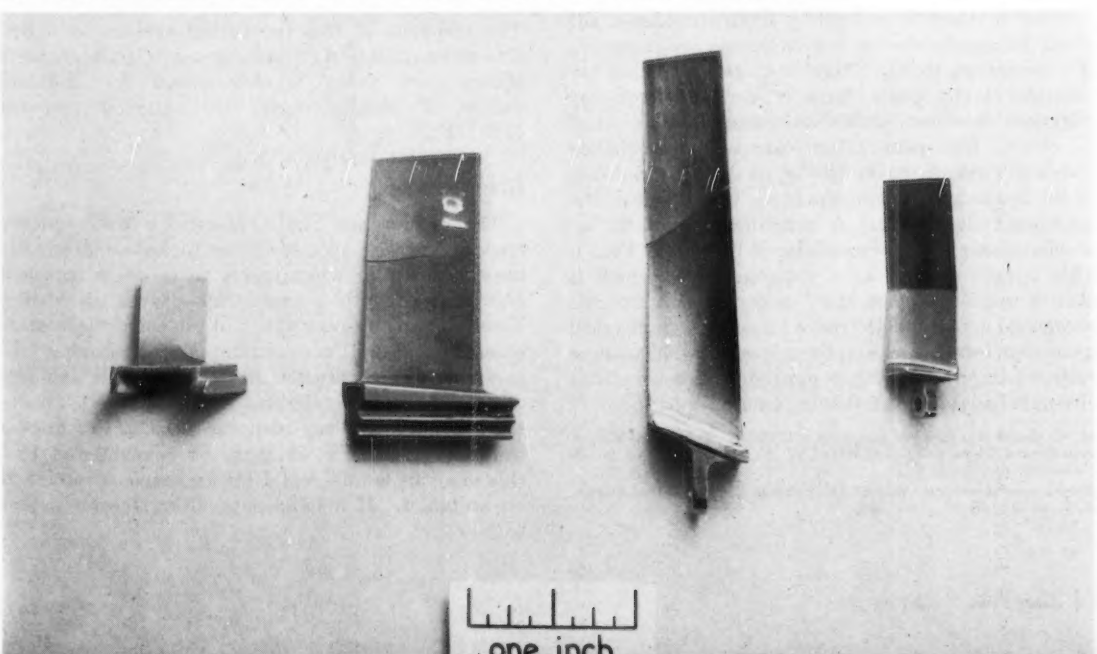
Fig. 5—Higher mode fatigue failures; frequencies (left to right) 9500, 5000, 1900 and 4540 cps



The Effect of Accelerometer Low-frequency Response on Transient Measurements

Response of measuring system to a half sine-wave pulse, a triangular-wave pulse and a square-wave pulse is determined for different values of dimensionless low-frequency response

by M. E. Gurtin

ABSTRACT—This report investigates the effect of accelerometer low-frequency response on transient acceleration measurements. The accelerometer is represented by an equivalent capacitance and the amplifier by an equivalent input resistance. The response of this system to a half sine-wave pulse, to a triangular-wave pulse and to a square-wave pulse is found.

Introduction

Crystal accelerometers are now widely used for transient measurements. The effect of accelerometer high-frequency response on transient measurements has been investigated by Levy and Kroll,[1] Welch,[2] Weiss,[3] Frankland[4] and Biot and Bisplinghoff.[5] These papers show the optimum value of damping to be between 0.4 and 0.7 of critical damping and that to obtain an accuracy of 5% of the peak acceleration in measuring acceleration pulses having the general characteristics of triangular or half sine-wave pulses, an accelerometer should have a natural period of less than about one-third of the duration of the acceleration pulse.

None of these investigators have considered the effect of accelerometer low-frequency response on the measured pulse. Accelerometers utilizing resistance strain gages have a d-c low-frequency response; however, crystal-type accelerometers are, in effect, high-pass filters whose low-frequency response depends upon the equivalent capacitance of the accelerometer and the input impedance of the associated electronics. A simplified circuit for an accelerometer and its amplifier is shown in Fig. 1. This circuit consists of a capacitor (C), which is also a coulomb generator, generating a specific charge (q) across its electrodes for any given physical measurements. The amplifier is considered to be a resistive impedance (R) to ground. This simplified circuit is based on the following assumptions:

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Paper was presented at the 1959 SESA Spring Meeting held in Washington, D. C., on May 20–22.

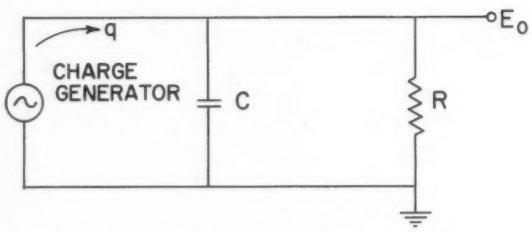

Fig. 1—Accelerometer equivalent circuit

1. The accelerometer high-frequency response is infinite.
2. There is no loading of the accelerometer and the amplifier due to the associated electronics.
3. The accelerometer may be represented by an equivalent capacitor (i.e., inductive and resistive effects are negligible).
4. The amplifier may be represented by a resistive impedance to ground.

The response of this measuring system to a half sine-wave pulse, a triangular-wave pulse, and a square-wave pulse is determined for different values of dimensionless low-frequency response ($\pi RC/T$).

Discussion

The results (see Fig. 2) show the low-frequency response of the accelerometer to be an extremely important factor when trying to measure transient accelerations with a reasonable degree of fidelity. To obtain an accuracy of 5% of the peak acceleration when measuring acceleration pulses having the general characteristics of the triangular or half sine wave pulses, an accelerometer must have a dimensionless low-frequency response ($\pi RC/T$) of greater than 20. However, it must be remembered that this analysis is only valid for an accelerometer and an amplifier. If a band-pass filter is used in the

system, the low-frequency response must be even lower. This is due to the fact that filters normally used in shock testing do not have a d-c low-frequency response. For a more detailed analysis of the effect of band-pass filters on transient measurements see Reference 6.

Analysis

Transfer Function of Accelerometer and Amplifier

The equivalent circuit of the accelerometer and amplifier is shown in Fig. 3. Using the Laplace Transform, the two-loop equations are written with all initial conditions zero.

$$\frac{i_2 - i_1}{CS} + i_2 R = 0 \tag{1}$$

$$i_1 = Sq \tag{2}$$

and

$$i_2 = \frac{E_0}{R} \tag{3}$$

Equations (1), (2) and (3) are combined to give the transfer function for the system

$$G(S) = \frac{E_0}{(q/C)} = \frac{S}{S + \frac{1}{RC}} \tag{4}$$

Transient Response of the System to a Half Sine-wave Pulse

The half sine-wave pulse is shown in Fig. 4.

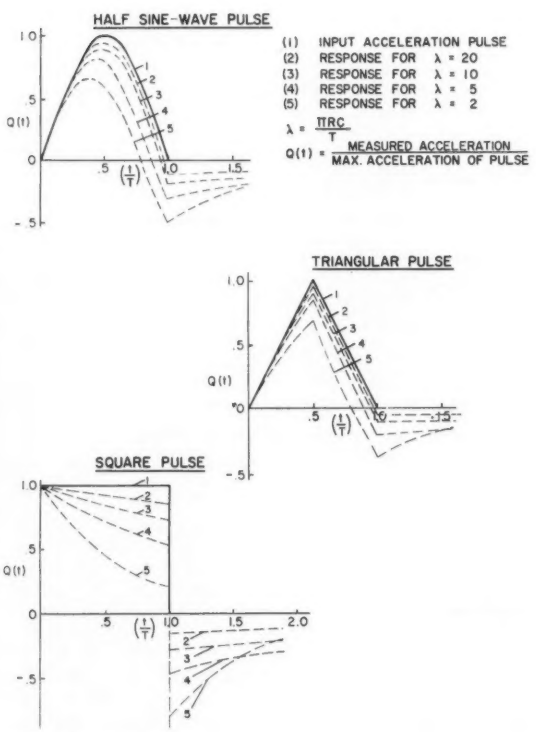
HALF SINE-WAVE PULSE
(1) INPUT ACCELERATION PULSE
(2) RESPONSE FOR λ = 20
(3) RESPONSE FOR λ = 10
(4) RESPONSE FOR λ = 5
(5) RESPONSE FOR λ = 2
λ = πRC/T
Q(t) = MEASURED ACCELERATION / MAX. ACCELERATION OF PULSE
TRIANGULAR PULSE
SQUARE PULSE
Q(t)
(1/T)

Fig. 2.—Results

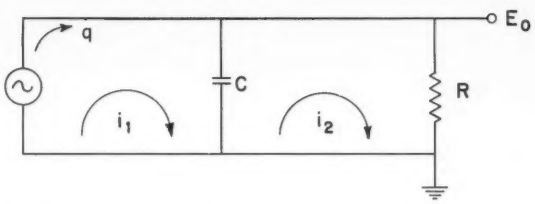

Fig. 3—Equivalent circuit

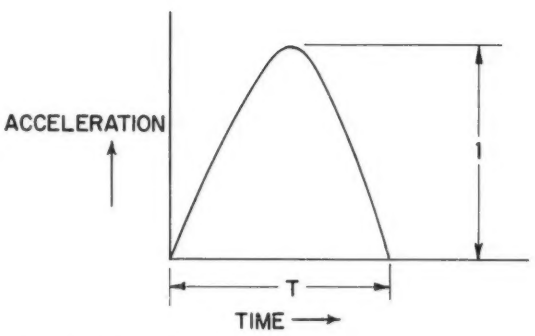

Fig. 4—Half sine-wave pulse

This input can be represented by the following equation:

$$I(t) = \sin \omega_0 t + \mu(t - T) \sin \omega_0 (t - T) \tag{5}$$

where

$$\mu(t - T) = \begin{cases} 0, & t < T \\ 1, & t \geq T \end{cases} = \text{a unit step function applied at time } T$$

The Laplace Transform* of (5) is taken

$$L[I(t)] = I(S) = \frac{\omega_0}{S^2 + \omega_0^2} [1 + e^{-TS}] \tag{6}$$

The output is given by the equation

$$Q(S) = G(S)I(S) \tag{7}$$

or

$$Q(S) = \left[\frac{S}{S + \frac{1}{RC}}\right] \left[\frac{\omega_0}{S^2 + \omega_0^2}\right] \left\{1 + e^{-TS}\right\} \tag{8}$$

If the inverse Laplace Transform of (8) is taken, there results

$$Q(t) = -\frac{\lambda}{1 + \lambda^2} e^{-\frac{\pi}{\lambda}\left(\frac{t}{T}\right)} + \sqrt{\frac{\lambda^2}{1 + \lambda^2}} \sin\left[\pi\left(\frac{t}{T}\right) + \psi\right] + \mu\left(\frac{t}{T} - 1\right)\left\{-\frac{\lambda}{1 + \lambda^2} e^{-\frac{\pi}{\lambda}\left(\frac{t}{T} - 1\right)} + \sqrt{\frac{\lambda^2}{1 + \lambda^2}} \sin\left[\pi\left(\frac{t}{T} - 1\right) + \psi\right]\right\} \tag{9}$$

where:

$$\lambda = \frac{\pi RC}{T} \tag{10}$$

$$\psi = \tan^{-1} \frac{1}{\lambda}$$

* *All Laplace transformations and inverse transformations were made with the aid of Reference 7.*

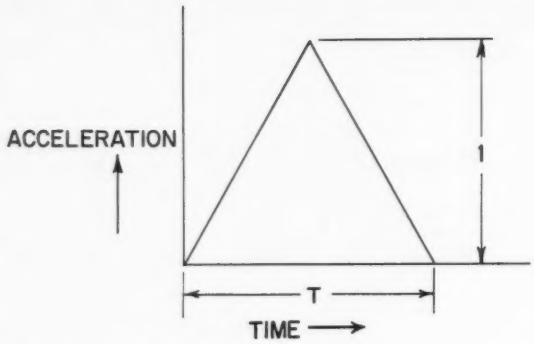

Fig. 5—Triangular pulse

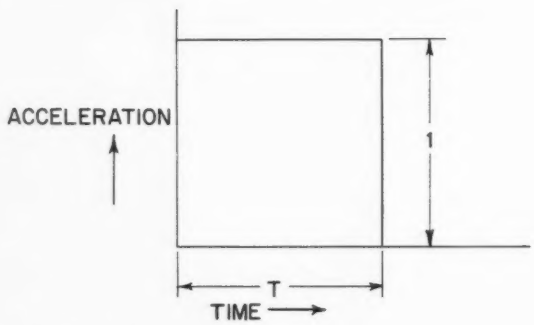

Fig. 6—Square pulse

Transient Response of the System to a Triangular Pulse

The triangular pulse is shown in Fig. 5. It may be represented by the following equation:

$$I(t) = \frac{2t}{T} - \mu\left(t - \frac{T}{2}\right)\left(\frac{4t}{T} - 2\right) + \mu(t - T)\left(\frac{2t}{T} - 2\right) \quad (11)$$

If the Laplace Transform of (11) is taken, the result is

$$I(S) = \frac{2}{TS^2} - \frac{4e^{-(T/2)S}}{TS^2} + \frac{2e^{-TS}}{TS^2} \quad (12)$$

The output then becomes

$$Q(S) = G(S)I(S) = \frac{2}{TS\left(S + \frac{1}{RC}\right)} - \frac{4e^{-(T/2)S}}{TS\left(S + \frac{1}{RC}\right)} + \frac{2e^{-TS}}{TS\left(S + \frac{1}{RC}\right)} \quad (13)$$

The inverse Laplace Transform of (13) yields the output as a function of time:

$$Q(t) = \frac{2\lambda}{\pi}\left[1 - e^{-\frac{\pi}{\lambda}\left(\frac{t}{T}\right)}\right] - \mu\left(\frac{t}{T} - \frac{1}{2}\right)\frac{4\lambda}{\pi}\left[1 - e^{-\frac{\pi}{\lambda}\left(\frac{t}{T} - \frac{1}{2}\right)}\right] + \mu\left(\frac{t}{T} - 1\right)\frac{2\lambda}{\pi}\left[1 - e^{-\frac{\pi}{\lambda}\left(\frac{t}{T} - 1\right)}\right] \quad (14)$$

Transient Response of the System to a Square-wave Pulse

The square pulse is shown in Fig. 6. It may be represented by the following equation

$$I(t) = \mu(t) - \mu(t - T) \quad (15)$$

Again the Laplace Transform is taken

$$I(S) = \frac{1}{S} - \frac{e^{-TS}}{S} \quad (16)$$

The output then becomes

$$Q(S) = G(S)I(S) = \frac{1}{\left(\frac{1}{RC} + S\right)}\left[1 - e^{-TS}\right] \quad (17)$$

The inverse Laplace Transform of (17) yields the output as a function of time:

$$Q(t) = e^{-\frac{\pi}{\lambda}\frac{t}{T}} - \mu\left(\frac{t}{T} - 1\right)e^{-\frac{\pi}{\lambda}\left(\frac{t}{T} - 1\right)} \quad (18)$$

Equations (9), (14) and (18) are plotted in Fig. 2 of this report.

Nomenclature

C	= equivalent capacitance of accelerometer.
E_0	= output voltage of amplifier.
G	= transfer function of accelerometer and amplifier.
i	= current.
I	= input acceleration pulse.
q	= charge generated by accelerometer.
Q	= output of measuring system.
R	= input impedance of amplifier.
S	= Laplace transform operator.
t	= time.
T	= duration of pulse.
$\mu(t - T)$	= unit step function applied at time T.
λ	= dimensionless low-frequency response = $\pi RC/T$.
ψ	= $\tan^{-1} 1/\lambda$.
ω_0	= fundamental frequency of half sine wave pulse = π/T.

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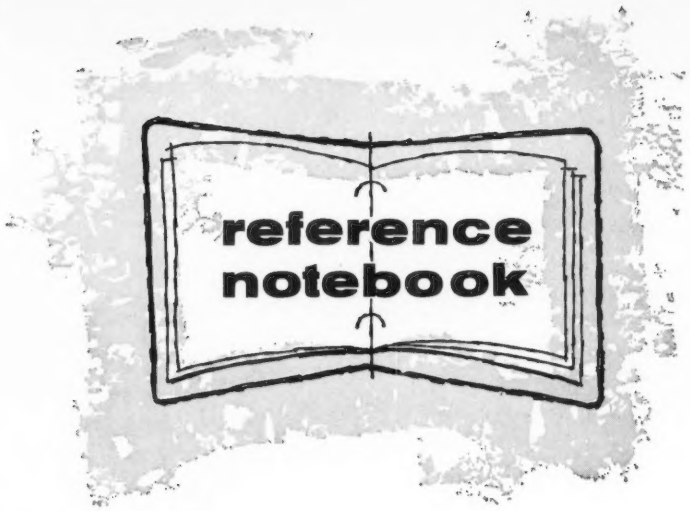

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1961 Exposition Stimulated by New Product Developments

As a concurrent feature of the First International Congress on Experimental Mechanics, the 1961 Exposition already has aroused considerable interest. It is being held this year on November 1–3 in New York City, at the Hotel New Yorker where the entire mezzanine floor will be conveniently arranged to accommodate all the major events of the Congress.

Reasons for the increased interest are found in the recent growth of research activity and of industrial applications in this expanding field. In response to the demand created by a tremendous growth, many product developments have made their appearance. In addition, new techniques and completely new products have opened experimental possibilities that did not exist before.

Economy-minded engineers and scientists quickly recognize the importance of accurate, time-saving devices and of new techniques that extend the scope of their investigations. Engineering management, also sensitive to the value of up-to-date equipment, is constantly searching for ways to save time and cut costs. The situation is more acute in the field of stress analysis and in the related fields of experimental mechanics where much of the work can be difficult and time consuming.

As a rule, the pressure of work is such that the engineer or scientist has little time to keep up in detail with all of the latest developments, although he may be aware, in a general way, of the improvements that are going on around him in his field. An actual show, such as the 1961 Exposition, serves the valuable purpose of bringing these men face-to-face with the newest developments in a relaxed atmosphere where manufacturers' representatives can demonstrate the qualities of their products.

The exhibition booths at the New Yorker are laid out around the main assembly hall so that manufacturers can display their products before the entire attendance. This year, the international scope of the Congress is expected to attract an especially large and interested audience.

Educational exhibits will also be a feature of the Exposition. They will consist of displays by universities, research organizations and governmental agencies that offer services, make new findings and produce new techniques of fundamental importance, but do not normally market products.

Particular interest has been aroused this year in a variety of products suitable for display. Among them are the following: All types of equipment used to impose static and dynamic loads on materials and components; testing machines; strain gages; photoelastic equipment and material; mechanical, electrical and electronic equipment and instrumentation for measuring, monitoring, reporting and recording the effects of loads; transducers; equipment used in producing and measuring force fields, accelerations, velocities, vibrations; equipment for simulating and measuring environmental extremes; photographic and X-ray equipment; and many others.

The 1961 Exposition is under the chairmanship of Theodore Braunstein of the Metropolitan New York Section. Working closely with him is Leonard Vidmark of SESA headquarters. Further information concerning the Congress and the Exposition will appear in future issues of EXPERIMENTAL MECHANICS.

Fourth U. S. National Congress of Applied Mechanics

The U. S. National Congresses of Applied Mechanics are held every fourth year under the auspices of the U. S. National Committee on Theoretical and Applied Mechanics, a joint committee of the American Institute of Chemical Engineers, the American Mathematical Society, the American Physical Society, the American Society of Civil Engineers, the American Society of Mechanical Engineers, the Institute of the Aerospace Sciences and the Society for Experimental Stress Analysis. The 1962 meeting will be at the University of California, Berkeley, June 18-21. These Congresses are planned to supplement the International Congresses of Applied Mechanics but not to compete with them. Accordingly, no effort is made to attract papers from outside the U. S. and Canada, although there is no rule against it.

All research workers in the field are cordially invited to submit papers constituting original experimental or theoretical contributions to applied mechanics, including mechanics of rigid bodies and deformable solids, mechanics of fluids and gases, thermodynamics and heat transfer. It is expected that papers accepted by the Editorial Committee with the advice of recognized authorities and presented at the Congress, will be published in full in the Proceedings of the Congress.

To be considered for presentation at the Congress, complete papers and manuscripts must be submitted to the Chairman of the Editorial Committee before Jan. 1, 1962; to be scheduled for presentation, the final manuscript of a paper must have been accepted before May 1, 1962. To avoid delays caused by overburdening reviewers and editorial staff, authors are urged to submit manuscripts well ahead of the deadline of Jan. 1, 1962.

The papers will be grouped by subject and 30 min will be allotted for presentation and discussion of each paper. Arrangements will also be made for general lectures by outstanding authorities on subjects of general interest to members of the Congress. Facilities will be provided for informal discussions and social contact.

Inquiries should be addressed to W. Goldsmith, Secretary, 4th U. S. National Congress of Applied Mechanics, University of California, Berkeley 4, Calif.

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section reports

Chicago

Nondestructive Testing

The March meeting of the *Chicago* Section was held at Magnaflux Corp., Chicago, Ill. After the meeting, personnel from Magnaflux Corp. presented a lecture and demonstration on various types of nondestructive testing. The demonstration included Magnaflux, Zyglo, eddy-current testing and the Elastomat machine (determining the modulus of elasticity of a material by the natural frequency of a specimen). A discussion period followed the lecture and demonstration.

Before the meeting, members and guests attended a dinner at Meo's Norwood House Restaurant.

(Reported by W. R. Klinke for the Chicago Section)

Milwaukee

Strain-gage Transducers

The March meeting of the *Milwaukee* Section was held at the ESM Building on Mar. 16, 1961. Forty members were present.

Warren C. Young, professor in the Mechanics Dept. of the University of Wisconsin, spoke on "The Design of Strain-gage Transducers." Professor Young is the originator and teacher of the graduate course in stress analysis techniques at the University.

His talk was a natural outgrowth of his course, and was effectively a continuation of it. He derived some equations which can be used in building special purpose transducers for private projects. In particular, he gave the most general forms of the equations to allow maximum versatility in application.

The next meeting was announced for Apr. 27, 1961, at the ESM Building.

(Reported by D. E. Niles for the Milwaukee Section)

Northern California

Structural Testing the X-15

The *Northern California* Section held its spring meeting on March 10, 1961, on the University of California campus at Berkeley. O. L. Gillette of North American Aviation, Incorporated, spoke on "High-temperature Structural Testing of the X-15." A very comprehensive talk, illustrated with many slides, covered the rapid heating systems, structural loading devices and the automatic instrumentation used. Highlight of the presentation was a movie showing the prototype in a speed run.

The evening was well spent by the forty people in attendance.

(Reported by D. M. Cunningham for the Northern California Section)

Northern Ohio

Stress Analysis in the Fastener Industry

The unique problems involved in the measurement of stresses on a nut-punching tool were very ably presented by W. Tuppeny at the meeting of the *Northern Ohio* Section on Feb. 16, 1961. Mr. Tuppeny's presentation was based on his experiences while at Lamson & Sessions and on work done under a grant of the Industrial Fasteners Institute at Case Institute of Technology.

Preceding the meeting, members and guests viewed the fastener manufacturing facilities of Lamson & Sessions in Cleveland, Ohio. Aside from the fact that fasteners present a stress-analysis challenge, the automatic machinery producing nuts and screws commands an analysis program of its own. Questions raised during

PANEL EXPERTS DISCUSS SCREW FASTENERS

Following a tour of the Lamson & Sessions Co. plant at Cleveland, Ohio, the February 16 meeting group of the Northern Ohio Section questions a panel of company experts. Facing the camera from left to right are: D. Brown, A. Breed, Section Vice-chairman Fred Main, L. Lesnak, R. Gaty, Section Chairman Howard Calvert, C. Harvey, W. Waltermire, Meeting Speaker W. Tuppeny and F. Monaco

the plant tour were answered by a panel of company experts introduced by Leo. F. Lesnak, personnel director and assistant to the vice president, manufacturing.

Stress analysis on a production machine is inhibited by the fact that any downtime is frowned upon by the production department. However, instrumentation and test time is always lengthened by newly uncovered problems. In Mr. Tuppeny's case the strain-gage clearance problem was so acute that any small protrusion of the installed gage caused its rapid destruction. The final setup consisted of a full Wheatstone bridge of foil gages waterproofed with a thin layer of ordinary fingernail polish. This novel form of waterproofing held up quite well, the speaker said, considering the violently changing tool temperatures encountered.

Mr. Tuppeny concluded the evening's presentation with several slides showing the stress-distribution patterns of photoelastic models of screw fasteners under load.

(Reported by N. Varju for the Northern Ohio Section)

Variety Panel Discussion

The April 11 meeting of the *Northern Ohio* Section was held in the Mechanical Engineering Building of Case Institute of Technology, Cleveland, Ohio. A panel of seven speakers gave talks on a variety of subjects in the field of testing. There were about 100 members in attendance.

Prior to the meeting a number of members enjoyed a tour through the ME building where new equipment and facilities were inspected.

The speakers and their topics were as follows: J. L. Biseai, Fanner Mfg. Co., "Impact Measurements in the Railroad Field"; R. L. Rubin, Lester Engineering, "Proving Rings based on SR-4 Strain Gages"; P. J. Chiarito, N. A. S. A., "Strain Measurements at Low Temperature"; R. W. Narter, Babcock & Wilcox Co. (Research Center), "Developments in High-temperature Experimental Stress Analysis Techniques by B & W"; H. H. Stroebel, Alcoa, "Determining Forces in Engine Rocker Arms"; T. S. Eickelberger, Thompson-Ramo-Wooldridge, "Temperature-indicating Paints"; S. Grosshandler, Performed Line Products, "Telemetry."

It was decided to hold the next meeting jointly with the Detroit Section at Toledo, Ohio, the date to be arranged later.

(Reported by H. Calvert for Northern Ohio Section)

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Southern Michigan

Nuclear Reactors—Principles, Engineering Problems and Safety Considerations

"Nuclear Reactors" was the subject on which C. C. Perry, professor at Wayne State University, spoke at the March 15 meeting of the *Southern Michigan* Section. The meeting, which was attended by approximately 35 members, was held at the Chrysler Corporation Training Center in Centerline, Mich.

In an informal but lively and entertaining manner, Professor Perry first instructed the audience on the basic principles of nuclear reactors. His discussion, ranging from the first exciting experiments in which new elements were created, to the design of "typical" reactors, provided a background for the further discussion of thermodynamic and structural engineering problems.

After outlining a number of the severe problems associated with "in reactor" strain measurement, Professor Perry elaborated the safety aspects of reactors. This final portion of his talk was augmented by an AEC film which emphasized the "built in" safety features of a boiling water reactor. The film showed a number of intentional "runaway" experiments, the last of which included an intentional steam explosion. The audience was particularly impressed with the safety features of reactors and the extremes to which one would have to go to even imagine the possibility of an accidental nuclear explosion.

(Reported by D. E. Martin for Southern Michigan Section)

Washington Area

"The time has come," the walrus said,
"to speak of many things.
"Of subs and ships and sealing wax, of
reinforcement rings,
"And why the sea is boiling hot, and
when do bugs grow wings."
(Apologies to Lewis Carroll)

That the rapid development of undersea craft is due in large measure to strain-gage technology, was disclosed on March 22 by Ralph B. Allnutt of the David Taylor Model Basin at one of the most interesting meetings in recent years of the *Washington Area* Section. Without strain gages and advanced sealing and waterproofing techniques, the complex stress patterns in submarines at the junctures of intersecting shells might still be unknown.

Mr. Allnutt's talk, "The Experimental Stress Analysis of Ships and Submarines," also covered the experimental solution of structural problems encountered in the *Forrestal* and other naval vessels. Using an extensive set of slides as illustrations, Mr. Allnutt recounted numerous projects in which strain gages were used to evaluate various types of reinforced openings in box girders, to study the capacity of older flight decks to handle heavy bombers, and so forth. Particularly noteworthy was a description of a facility for the fatigue testing of pressure vessels, using force-time histories recorded in actual service. In discussing the results of investigations into the brittle fracture problem, Mr. Allnutt, who is the head of the Model Basin's Performance Evaluation Branch, revealed that many ships still in service were fabricated with steels which are susceptible to brittle fracture in rough seas.

A lively discussion followed the talk, after which a fascinating film, "The Rival World," was shown. Produced by the Shell Oil Co., the film was a documentary on insects, dealing chiefly with the life cycle of the locust and mankind's war against it in Africa.

(Reported by L. Mordfin for the Washington Area Section)

Denver Group Holds First Meeting

The first organizational meeting for the purpose of starting a local section was held on April 4th by the Denver, Colorado, group. Although the weather was bad, 26 persons turned out to hear SESA Local Sections Chairman Clarence R. Smith of Convair, San Diego, Calif., talk on "The cause and Prevention of Fatigue Failures in Metal Parts."

A steering committee of four was appointed to continue organizational work and to prepare for another meeting during the last week in May. Serving as representative for the area is M. H. Barrett of Ketchum, Konkel & Hastings, Consulting Engineers, 730 Kalamath St., Denver 4, Colorado. The meeting was described as "excellent".

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STAFF COMPLETED FOR SESA-UCLA COURSES

Announcement was made recently of the completed staff of instructors who will present the three short courses on experimental stress analysis at the University of California at Los Angeles this summer. Given under the joint sponsorship of the SESA and Engineering Extension, Department of Engineering, UCLA, the three courses comprise: Strain Gage Lectures—August 21–25; Strain Gage Laboratory—August 28–September 1 and Industrial Photoelasticity—September 5–9.

The staff members who will participate in the program are: William T. Bean, consulting engineer, Detroit, Mich.; Ward B. Brewer, Engineering Research Dept., Lockheed Aircraft Corp., Burbank, Calif.; Jerome Catz, Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass.; Edward W. Larson, Jr., Rocketdyne Div., North American Aviation, Inc., Canoga Park, Calif.; David J. Leeds, Department of Engineering, University of California, Los Angeles, Calif.; Milton M. Leven, Research Laboratories, Westinghouse Electric Corp., East Pittsburgh, Pa.; William M. Murray, Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass.; Joseph C. Sanchez, Micro Systems, Inc., San Gabriel, Calif.; Clarence R. Smith, Structures and Materials Laboratories, Convair, San Diego, Calif.; Peter K. Stein, Stein Engineering Services, Inc., Phoenix, Ariz.; Leon J. Weymouth, Electronics and Instrumentation Div., Baldwin-Lima-Hamilton Corp., Waltham, Mass.; Felix Zandman, Instruments Div., The Budd Co., Phoenixville, Pa.

STRAIN-GAGE INSTALLATION KIT

Display of strain gages and installation equipment that comprises kit furnished registrants during the 1960 strain-gage courses at MIT and UCLA.

RESEARCH AT CASE

Progress on the frontiers of scientific knowledge was the basis for Case Institute of Technology's Twelfth Annual Research Day attended by 200 leading Ohio industrialists and research scientists.

The day-long program featured discussions by the Case faculty and graduate students of their investigations in new areas of research, ranging from plasma physics (which reveals hitherto unknown sources of energy) and the chemistry of matter in the outer reaches of space. Highlight of the day's activities was a banquet address by Case President, T. Keith Glennan.

Research at Case, which can trace its inception to Albert Michelson, Case's first instructor in physics and America's first Nobel Prize winner in science, has advanced rapidly. The current research budget is $2.5 million a year.

Seventeen research subjects, which were discussed during four concurrent morning sessions, examined research investigations in the fields of chemistry, metallurgy, physics and engineering. Following a luncheon in Tomlinson Hall, the guests met in Strosacker Auditorium for a general session concerned with topics of interest to all areas of scientific research.

The afternoon program included discussions of Case's behavioral science group, digital control systems engineering and tours of the new multimillion dollar facilities now available on the campus. Robert Thomas, director of research administration, served as chairman of the afternoon session.

VAN ARSDALE BECOMES PRESIDENT OF BRADLEY

Talman W. Van Arsdale, Jr., Worcester Polytechnic Institute's executive vice president, has been named president of Bradley University, Peoria, Ill.

Dr. Van Arsdale will take over as Bradley's sixth president on July 1. He succeeds Harold P. Rodes, now president of General Motors Institute, Flint, Mich.

Bradley, founded in 1897, has 4500 students in 11 colleges, schools and institutes. Like Worcester Tech, it is independent and nontax supported. WPI, founded in 1865, has an enrollment of 1150.

Dr. Van Arsdale has been associated with Worcester Tech since April 1956. He served first as vice president for development and public relations, moving up the next year to become the college's first executive vice president.

Three Short Intensive Courses on

STRAIN GAGES & PHOTOELASTICITY

at the DEPARTMENT OF ENGINEERING
UNIVERSITY OF CALIFORNIA AT LOS ANGELES

FOR WHOM INTENDED

These three courses have been designed to present an integrated program on theory and practice, including a review of fundamentals and discussions of some of the latest techniques, especially those suited to high and low temperature strain gage applications. Attention will also be given to instrumentation, which, in many cases, can be used equally well with other kinds of transducers.

The courses are intended primarily for those with a bachelor's (or higher) degree in engineering, physics, or metallurgy. However, in certain cases, experienced technicians without degrees can be expected to gain full benefit.

The material content of the courses, although presented largely from the point of view of the mechanical engineer, has been selected to provide maximum benefit for those who are already skilled in stress analysis but who may have little experience in electrical instrumentation, and likewise for electronics experts who may not be quite so familiar with certain aspects of stress analysis.

STRAIN GAGE LECTURE COURSE

August 21 - 25, 1961 (Monday through Friday)

The first of the two strain gage courses will consist of a series of lectures supplemented with illustrative demonstrations. Some of these lectures have been prepared especially for beginners and consequently are of an elementary nature; however, other lectures of a much more advanced character will also be included in the series. The latter group will contain detailed information, especially relating to high temperature strain gages, which will be useful to those who already possess considerable experience. The content of the course has been selected and organized for maximum benefit to designers, metallurgists, laboratory workers, laboratory supervisors, theoretical stress analysts, and others, who wish to advance their knowledge relating to the uses of strain gages and the techniques which are available for producing optimum results.

Both the theoretical and practical aspects of strain gages themselves, methods of installation, and allied equipment, will be presented in the lectures. Details of strain gage applications for measuring force, torque, bending, shear, and accelerations, will be discussed as well as simple electric circuits for automatic computation of quantities which depend upon the simultaneous observation of strains at more than one location.

The following will be included in the topics which are to be presented:

Basic theory of Wire, Foil, and Semi-Conductor strain gages
Details of gage characteristics
Techniques for installing gages
Mechanical and Electrical aspects of the gage system
Fundamental Laws of Measurement
Electric Circuits for strain gages
Dynamic and Static Strain Measurements
Computing Bridge circuits for direct measurement of quantities depending upon several strains
Stresscoat as an Auxiliary Tool
Measuring systems and instruments
Low-temperature applications of strain gages
High-temperature strain gages and their installation
Commercial Equipment
Rosette analysis and interpretation of observations

STRAIN GAGE LABORATORY COURSE

August 28 - September 1, 1961 (Monday through Friday)

For those who wish to gain personal experience in the installation of gages and the handling of equipment, a second course containing a series of laboratory experiments will be offered. The laboratory program is supplementary to the lecture course. It will consist of a set of projects, each containing one or more experiments, which have been designed to illustrate the topics taken up in the lecture course during the previous week. The following exercises will be included to provide practice in the practical applications of strain gage techniques:

Strain Gage Installation for normal and high temperature applications
Static observations with single and multiple gage bridges
Dynamic strain Measurements
Characteristics and checking of electrical equipment used with strain gages
Strain Rosette Analysis
Characteristics and handling of certain commercial equipment
Measurement of Force, Torque, Bending, and Moments at inaccessible locations

Since the benefits which can be obtained by attendance at the laboratory program depend to a large extent upon an understanding of the underlying theory, registration in the laboratory course will be limited to those who have attended the previous lecture course, or its equivalent in prior years.

INDUSTRIAL PHOTOELASTICITY

(Lectures & Laboratory)

September 5 - 9, 1961 (Tuesday through Saturday)

The course in Industrial Photoelasticity will review the fundamental theory and laboratory techniques related to the classical applications of polarized light for solving stress problems. In addition to this, discussions of some of the most advanced techniques, including Three-Dimensional Photoelasticity and Photoelastic Surface Coatings (PhotoStress), will also be presented.

The course will emphasize the ways in which photoelasticity can be used to solve industrial problems. Attention will also be given to the advantages and limitations of the method and its relation to other experimental techniques with which it may often be combined most effectively.

The material will be presented through lectures, demonstrations, and laboratory exercises. Consideration will also be given to the properties and selection of photoelastic materials as well as the specification of equipment required to set up a laboratory. This program has been designed especially for the engineer or metallurgist who has little or no previous knowledge of the subject.

TUITION

The tuition per registrant for each course will be as follows:—

Strain Gage Lecture Course	**$175.00**
Strain Gage Laboratory Course	**$175.00**
Industrial Photoelasticity	**$200.00**

FOR FURTHER INFORMATION, call or write,

ENGINEERING EXTENSION, DEPARTMENT OF ENGINEERING,
Room 6266, ENGINEERING-PHYSICAL SCIENCES BUILDING,
UNIVERSITY OF CALIFORNIA, LOS ANGELES, 24, CALIFORNIA

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